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SOUNDPROOFING AND FILLING IN OF ENGINES

J. Rauch

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These acoustic resonances cause filling in fluctuations which give rise to various disadvantages. The knowledge of their causes allows removal of those which turn out to be harmful and reinforcing those which are capable of improving qualitatively and quantitatively the filling in curves by optimizing their shape. There are good grounds for adding the two conditions of flow and three conditions of filling in to the four additions of soundproofing and carburation.

PREFACE[1]

The problem of noise emitted by vehicles is a present day problem. It is certainly possible to produce a suitable attenuation of these noises, but it is often only at the cost of expensive or awkward installations. From this standpoint, special attention should be brought to those factors which cause periodic flow of gaseous streams passing across any alternating engine and which give rise to intake and exhaust noises.

It has been known for some time that a suitable arrangement of sound attenuation devices can have a favorable effect on the operating of a machine. Nevertheless, since it is always a question of periodic phenomena, the advantage only becomes clear in the case of some favored rotational speeds (or specific speed areas). A proper application of conventional theories in the acoustics and mechanics of fluids should allow specification of these phenomena and isolation of the effect of fundamental parameters. This has been the special work of M. J. Rauch, a work which has led him to suggest extremely interesting rational solutions which have been applied, moreover, successfully in actual design.

In the first part, the author reviews some conventional acoustical data relating to sound waves in pipes and volumes as well as describing the chief resources available to attenuate their intensity. Used in mufflers, these attenuation resources lead to the production of acoustic filters (similar to electrical filters) whose advantages and disadvantages are quite clear. The second part deals with the soundproofing of the exhaust and the third part with the soundproofing of the intake. In this last case, the requirements for a good carburation have not been forgotten. The study of filling of piston engines is examined in the last part. The harmonic analysis of the movement of a gaseous flux in the intake pipe causes the appearance of pressure waves capable of exciting this pipe in resonance. The aerodynamic disturbances which result well explain the presence of "holes" of power for specific rpm of the

[1]R. Comolet, Professor at the Sorbonne.

engine, but at the same time suggests ways of suppressing them. This is certainly the most original part of the author's investigations.

The conclusions of this work should allow engineers involved with alternating engines to acquire some degree of mastery in the design of shapes and arrangements of input and exhaust pipes, a mastery which will not be owing to the application of a skillful empiricism but to a better theoretical knowledge of the phenomena and their interplay. We are grateful to M. J. Rauch for having cast some light on these difficult problems whose solutions generally lie undisturbed in the files of research offices.

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SUMMARY

There is a close connection between filling in and soundproofing of the exhaust and intake of piston engines. Both these problems cannot be separated for they are linked with the very concept itself of muffler devices. This concept should take into consideration the requirements of soundproofing on one hand and filling in on the other.

Poorly soundproofed, the exhaust and intake of the automobile piston engine form sources of extremely intense noise. Their soundproofing poses a certain number of problems in common and requires the application of identical acoustical principles of attenuation (reflection, absorption and even interference). The predominance of low pitch components in the noise spectra makes the problem of reduction of sound levels especially difficult because the systems of low frequency attenuation are the most difficult to establish.

The problems which are presented in the case of the exhaust are not equal in difficulty in the various arrangements of the one or more mufflers placed on the exhaust pipe in its true sense. The goal of this study has been to investigate solutions in the case of the arrangement of the least favorable main muffler, in the vicinity of the exhaust outlet. The best high pass is in this case the quarter wave resonator. When there is only one muffler based on reflection principles, an especially difficult problem arises which is that concerning suppression of the cut-out noise whose dominant frequency is a harmonic of a rather high order of the fundamental frequency.

The only attenuation of the intake noise which can be carried forward until total suppression of the latter is achieved calls upon the same acoustical principles. Although, in the case of the exhaust, it is a matter of chiefly dealing with resonances of pipes, the soundproofing of the intake is characterized in sum by the requirement for suppressing volume resonances, as much to prevent them from appearing in the form of noise as to avoid a related phenomenon of disturbance of the carburation in the case of external combustion engines. It is shown that 4 conditions are necessary for the intake muffler to be free of these defects without, however, removing entirely the intake noise. The total suppression of the latter can only be obtained with more complex mufflers, chiefly those containing a quarter wave high pass.

The initial goal of the filling in step was to succeed in suppressing the disturbances of filling in curves well known to automobile drivers under the term of "filling in holes". Two phenomena become superposed during the intake time which arise owing to the flow in its true sense with losses of load on one hand and with the effect of inertia of a gaseous column contained in the intake duct on the other hand. The latter gives rise to a modulation of the former and, beyond any resonance state, the actual pressure at the valve heads appears as the product of the pressure connected with bringing to speed beginning from the atmospheric pressure by a modulation factor with two harmonics. The existence of harmonics of the actual pressure and of those of the intake duct considered as a quarter-wave pipe allows explanation of the shape of a filling in curve. The latter results from the amplification and the modulation on a base curve, disregarding resonance phenomena, by acoustic resonances which form the origin of what can be called acoustic supercharging.

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SOUNDPROOFING AND FILLING IN OF ENGINES

J. Rauch[1]

ABSTRACT: The acoustic principles are presented for attenuating automobile engine noise. The various aspects of soundproofing exhaust noise are discussed, along with the means for low pitch sound absorption; and a terminal reflection muffler system for total noise suppression is described. An admittance type muffler is also considered, and the requisite dynamic and acoustic conditions for its use are determined. Different materials are proposed for use as sound absorbers; and the case of acoustic supercharging and optimization is studied.

INTRODUCTION

Our first intention was to limit the subject of this work to the filling /1* in in its true sense of piston engines. But, for practical purposes, the report would have been tedious and undoubtedly too incomplete. It appeared preferable for us to reveal the close link existing between filling in and soundproofing of the exhaust and intake. In reality, these two concepts are closely related to the muffler concept itself. This concept takes into account the requirements of soundproofing, on one hand, and filling in, on the other. It is also believed that there are two essential problems presented in any study of intake and exhaust of a piston engine: an acoustical problem and a flow problem.

In the interest of clearity, we shall first of all describe the acoustical principles employed in the exhaust and intake depths as well as in the mufflers attached to them. There is defined (Part 1), in the case of the attenuation systems planned, a characteristic attenuation which lends itself to a quantitative study (reflection, absorption, interference) and a total

[1]Engineer of Arts and Manufacture.

*Numbers in the margin indicate pagination in the foreign text.

attenuation whose qualitative aspect is practically the only one which could be taken into consideration.

The special problems posed by exhaust and intake in so far as concerns their soundproofing forms the subject of two separate parts (2nd and 3rd), where the accent is placed on the most difficult questions.

When the space available for the exhaust is limited, the most difficult soundproofing problems are the attenuation of the low-pitched sounds and suppression of the cut-out noise.

The study of the intake should cope with similar problems and end up logically with a muffler pipe which should satisfy six conditions of flow, soundproofing, and carburetion.

Discussion of the problem of filling in begins in Part 4. The problem itself if overshadowed by phenomena of acoustic supercharging (intake) and acoustic counterpressure (exhaust) subjects we have brought forward and discussed in detail. Already anticipated in some fixed installations (Brown and Boveri), the acoustical phenomena of intake ducts have not been exploited on a large scale, with an engine whose speed was essentially variable like that of an automobile, before the appearance of the Peugeot 404. Some essential problems had to be resolved, chiefly the suppression of all disturbance of the filling in curve, which required a considerable amount of laboratory time. It was possible to eliminate the "filling in holes" and it is now also possible to optimize the shapes of the filling in curves of an engine. It has been shown that, in addition to the above-mentioned six conditions, there are three supplementary conditions concerning the filling in insofar as the intake muffler is concerned and that three conditions should be complied with in the design of the exhaust.

This experimental study was carried out at the Peugeot automobile company on the 403 and 404 engines, the former being used above all for study of soundproofing problems of the exhaust and intake, the latter for study of the filling in problem. /2

I should like to express my deep appreciation to M. R. Comolet, Professor at the Faculty of Sciences of Paris for his interest in these investigations as well as the valuable advice and encouragement given me.

I should also like to thank Messrs. R. Lucas, Director of the Advanced School of Physics and Chemistry and R. Vichnivsky, Professor at the Faculty of Sciences of Paris.

I also express my gratitude to Messrs. M. Dangauthier, Director of the Center for Research, Investigations and Testing of the Peugeot automobile company and L. Hogard, Chief Engineer at the Peugeot automobile company. I should also like to thank Chief Engineer T. Vernotte without whom this publication would not have been possible./

PART 1
ACOUSTIC PRINCIPLES

CHAPTER I
SOUNDPROOFING IN THE AUTOMOBILE

General Comments

The present degree of perfection of the automobile is the fruit of patient /5
toil in all fields concerning it. This was possible:

1. For the mechanics in its true sense, owing to a continuous increase in the specific power and to the search for increasingly rational solutions as much for the engine itself as for the transmission and suspension, together with a permanent concern for weight reduction;

2. In the case of the body, owing to the development of new design principles better and better satisfying the fourfold requirement for habitability, aerodynamics, lowering of the center of the gravity and reduction in weight;

3. Insofar as concerns the two systems, owing to the development and to the application on an industrial scale of a completely new technique, soundproofing, a product of the unavoidable consequences, of an acoustical and physiological order, of the continuous tendency for increase in the weight/ /power ratio.

The goal of soundproofing is to lower, as much as possible, the sound levels generally and to attenuate the different components of noise to the extent of the special requirements for each of them, taking into account physiological realities whose description has been made chiefly in the network of curves of equal sound sensations produced by Fletcher and Munson.

This definition of soundproofing is applied to any problem of attenuation of total noise and when it is considered that in the case of a machine as complex as a modern automobile, for example, this total level is a result of different noises with very diverse origins, it appears that the acoustical laws themselves are hardly of a nature to simplify the task of the engineer.

Very generally, soundproofing becomes more difficult the lighter the system. Two methods of soundproofing can be discriminated:

a. The one involving action on the sources of noise;

b. The one involving action on the transmission of noise to the inside and to the outside of the body.

This transmission is done, either by mechanical (mechanical links), or through the air, or by both of these at the same time.

Both above mentioned methods are often carried out simultaneously. As /6 for means of soundproofing within the wider meaning of the term, we shall confine ourselves to list them after having described the general process of soundproofing a new automobile:

1. Soundproofing at the system's design stage;

2. Soundproofing at the design stage of the different parts;

3. Soundproofing after the fact of systems, parts and main components.

These three stages involve use, in the most general case, of a combination of the following attenuation modes: damping, absorption, differentiation of frequencies (of excitation and resonance), mechanical filters, acoustic filters, etc.

Sources of Noise

In the automobile, the low-pitched components come from the forces of inertia of the moving parts of the engine, the exhaust, the intake, the transmission, the suspension, the rolling and the penetration of the body into the air. Their incidence is greater when their transmission to the inside of the body can be facilitated by the entering into resonance of certain components of the body frame and including the volume of air contained in the latter.

The sources of high-pitched components are the engine distributor, intake, exhaust, transmission, suspension, noises from rolling and aerodynamics.

The exhaust and intake therefore only represent two sources of noise among many others.

Rule for Addition of Decibels

Let us assume that the sound level D in decibels is to be lowered. This sound level is the result of n + 1 components $D_0 \geqslant D_1 \geqslant D_2 \ldots \geqslant D_n$ in decibels. When ρ_i is the ratio of the energies of components D_0 and D_i, the total resultant level D will have the value in decibels:

$$D = D_0 + 10 \log\left(1 + \frac{1}{\rho_1} + \frac{1}{\rho_2} + \ldots + \frac{1}{\rho_n}\right).$$

It results from this that even a large attenuation, of one of the above mentioned components, D_1 for example, will only lead to a slight drop in the total initial level as long as D_0 has not been also corrected at least to D_2. When D_0 and D_1 have each been corrected to D_3, for example, it will become required to act upon D_2 before any other thing, etc. Soundproofing is therefore to be considered as a typical example of teamwork. D_0 can represent a noise of engine suspension, D_1 an exhaust noise, D_2 an intake noise, D_3 a fan noise, D_4 a transmission noise, etc. Let us now show that, as far as the problem of reduction of noise level is concerned, the direct consequences of the law of addition of decibels can appear confusing. Let us assume that we /7 are tasked with suppressing one of the most energetic components of a total noise to be attenuated. The result of its suppression will be ever so much more spectacular when the aforementioned component exceeds the others by a more impressive number of decibels and when the other components are few in number. Nevertheless, the reciprocal can also be anticipated: even when we assume the case of the existence of a single other component, if this latter has initially the same value as the one that is to be suppressed, there can be no hope for gaining more than 3 dB out of the total noise at the maximum, and even so this slight gain does assume the total suppression of the component under consideration.

However, things appear differently when it is planned to act on the composition of the noise. The almost total suppression of some components can become a requirement when the latter are deemed unacceptable. The human ear is a rather good noise analyzer and physiologically a given noise is ever so much more unbearable when it contains frequencies radically differing from one another.

Be that as it may, the program for any soundproofing study should take into account, in addition to the above data, the following fact: considered separately, the different components do not have the same repercussion on the ear given equal energy. The well-known curves of equal sound sensations of Fletcher and Munson provide a complete idea of them.

Another goal to be pursued in soundproofing is the suppression of any "noise peak": when a uniformly increasing level, for example, as a function of the operational mode of the engine or speed of the vehicle, is rather well tolerated, any transition of the sound level through a maximum value for a relatively restricted variation of the operational mode or "noise peak", is to be avoided.

Position of the Problem

After this rather pessimistic view of the soundproofing problem, let us try to make ourselves an idea of the resources that could be planned for use in successfully attenuating two sources of noise which will be taken under study within the scope of this work.

It appeared absolutely necessary for us to limit this work to a presentation of contributions which are effective within the fields of intake and exhaust of piston engines. It is therefore almost superfluous to specify that, just as in any study of this kind, it will only be possible to review those different aspects of the questions treated which are already well-known and have been studied. On the other hand, the accent will be placed on new observations as well as on some problems which have as yet had no solution and which we propose to solve. The position of the problem of soundproofing the exhaust and intake can be described as follows:

Let there be, in a pipe with a section S, p the acoustic pressure, ρ the specific gravity of the gas, u the laboratory speed of the plates, also called the particular speed, c the speed of sound. The definition, by electro-acoustical analogy, of the characteristic impedance is:

$$Z = \frac{p}{Su} = \frac{\rho c}{S} .$$

Let us recall that, in a progressive plane wave, Z is an actual quantity which is related to an acoustical resistance. In the contrary case, when there are reflections, the impedance includes imaginary terms corresponding to the electrical reactance. Z is actual when p and n are in phase or in phase opposition. In the contrary case, Z is complex. /8

In the more general case, the impedance of a given pipe is to be considered as prescribed. Knowing in advance that it cannot be modified for reasons of size or room available, it is of little importance to know the exact value. One single thing will be of the utmost importance, and that is to know the attenuation in decibels that can be produced either from a total initial level or from the different components of the sound spectrum in question using acoustic filters. The advantage of calculating the aforementioned attenuations by component in decibels is therefore clear.

Before summarizing attenuation resources and reductions in decibels that can be anticipated using different filter principles, it is necessary to specify some fundamental acoustic properties of the basic components of the systems which will be studied: pipes, capacities and combinations of both, insofar as concerns their qualitative acoustic behavior.

<u>Resonance Modes of Pipes</u>

When there is resonance, the wave or soundwaves are said to be stationary. With any pressure wave (acoustic overpressure with respect to the static pressure of the environment) corresponds a wave with special speed (of elongation of sections) which is associated with it. Both these waves are phase shifted with respect to each other by $\pi/2$ or even by $\lambda/1$, λ being the wavelength under consideration.

A pipe with two free ends can be the seat of acoustic resonance phenomena whose fundamental vibration mode is as a half-wave. It follows that, under these conditions, when f_1 is the frequency of the wave $\lambda_1 = 2L$. L is the length of the aforementioned pipe and c is the speed of sound: $f_1 = c/2L$. All of the integral multiples of this harmonic 1 or fundamental will be able to occur on the list of resonance frequencies of such a pipe. It will then be possible to have in all, isolatedly or simultaneously:

$$f_1 = \frac{c}{2L},\ f_2 = \frac{c}{L},\ f_3 = \frac{3c}{2L},\ \ldots\ f_n = \frac{nc}{2L}\ .$$

Likewise, a pipe closed at one end and opened at the other can be the seat of isolated or simultaneous resonances at the fundamental frequencies of vibration as a quarter-wave and uneven harmonics:

$$f_1 = \frac{c}{2L},\ f_2 = \frac{3c}{1L},\ \ldots\ f_n = \frac{(2n-\ 1)c}{1L}\ .$$

Taking into account the characteristic dimensions of the pipes, it is possible to disregard transversed resonance modes (wavelengths on the same order as the diameter of the pipe).

Resonance Mode of Volumes

Although the qualitative and quantitative behavior of the pipes mentioned above can be described by different conventional calculating methods, the one for capacities is more directly performed through thermodynamics. For reasons of clearness there will first of all be taken up the resonance frequency of the resonator with volume or the Helmholtz with throat, i.e., the volume associated with a pipe (Plate 1 and Figure 5). It involves a volume V which plays the role of a spring in the corresponding mechanical diagram, and a throat l of section πr^2 which is associated with it. Such a system has a single resonance frequency on conditions that the wavelength of the exciting wave is much greater than the diameter of the sphere /9

$$f_0 = \frac{c}{2\pi}\sqrt{\frac{\pi r^2}{V(l + \pi/2r)}}\ ,$$

the term $\pi/2r$, not appearing directly in the calculation, is from Rayleigh and takes into account the fact that $\rho\pi r^2 l$ is not all the mass to be considered but that there are grounds for increasing it by two small end masses of the throat. The calculation mass slightly exceeds the geometrical boundaries of the throat. $\pi/2r$ is the correction for ends.

The nomogram of Plate 1 provides the resonance frequencies of such a system for the common geometrical values concerning us. It will be noted that in the case of T temperatures different from T = 288°K, there are grounds for multiplying f_0 by $\sqrt{T'/T}$, the speed of sound being

$$c = \sqrt{\gamma gRT} .$$

In the system MK_pS there is furthermore $\gamma = 1.11$ for air and 1.25 for the exhaust gases, and R = 29.3 as much for the air as for the exhaust gases.

When l is 0, the resultant system is reduced to the simple volume with opening of section πr^2, i.e., with the resonator with volume without throat. Its resonance frequency is:

$$f_0 = \frac{c}{2\pi}\sqrt{\frac{2r}{V}} .$$

Simultaneous Resonances of Pipes and Volumes

When the conditions of excitation (which we shall review later) are fulfilled, the throat l of the Helmholtz resonator can itself be the seat of phenomena of pipe resonances. Depending on the cases (which can occur and will be specified) the end of the throat opposite to that which empties into volume V is to be considered as open or closed. This distinction does not become a factor in the case of the Helmholtz resonator in its true sense, for which the aforementioned end is in principle subjected to a variation of pressure at the resonance mode.

When l is 0 and when the shape of V deviates perceptibly from a sphere, the qualitative behavior of the system becomes difficult to determine. This special case will not be taken into consideration in the following.

Comments

1. If with resonance the reflection at the open ends of the pipes were /10
perfect and if the nodes of pressure were located exactly in the planes of the sections of these ends, no sound energy would be transmitted to the outside.

In reality, there are grounds for taking into account an end correction as in the case of the throats of the Helmholtz resonators. Added to the developed length of the pipe considered, it provides its corrected length or by calculation, and is worth perceptibly 0.61r per completely free open end (Levine and Schwinger) and 0.8r when the latter empties into a volume (Rayleigh).

2. The phase shift of $\lambda/4$ of the pressure wave and special speed which exists at resonancy in the pipes, is no longer verified in the absence of reflection (plane wave progressive in an undefined pipe). An undefined pipe is to be considered as perfectly absorbent. Pressure p and special speed u are in this case constantly in phase along the pipe.

Let there be a source of shift σ of elongation $\sigma = \alpha \sin \omega t$ at the end of a pipe open at the other end (Figure 1); the wave takes time x/c to arrive at point M located at x from end σ of the pipe. At point M, there will be:

$$\sigma = \alpha \sin \omega\left(t - \frac{x}{c}\right) ,$$

and for the speed:

$$u = \alpha\omega \cos \omega\left(t - \frac{x}{c}\right) .$$

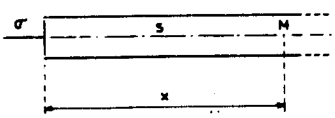

Figure 1.

Let us consider a section dx of mass ρSdx. It is possible to write:

$$\rho\left.\frac{\partial^2\sigma}{\partial f^2}\right| = \frac{\partial p}{\partial x} ,$$

where:

$$\left.\frac{\partial^2\sigma}{\partial f^2}\right| = -\alpha\omega^2 \sin \omega\left(t - \frac{x}{c}\right)$$

hence:

$$\Delta p = -\alpha\rho\omega \int_0^1 \sin \omega\left(t - \frac{x}{c}\right)dx$$

or again:

$$p_1 = \alpha\rho\omega c \cos \omega\left(t - \frac{x}{c}\right) + p_0 .$$

The acoustic pressure p is equal to $p_1 - p_0$, p_0 being the static pressure /11
of the environment:

$$p = \alpha\rho\omega c \cos \omega\left(t - \frac{x}{c}\right).$$

p and u are therefore in phase, and the characteristic impedance of the pipe with section S:

$$Z = \frac{p}{Su} = \frac{\rho c}{S}$$

is a pure resistance. Furthermore, no matter what the abcissa may be, this impedance is always the same towards the right hand in the case of Figure 1.

Levels and Noise Spectra

The various systems which could be considered are far from relating to the/ same acoustic plan. Nevertheless, it is clear that, no matter what this plan may be, it will include an amplification of excitation components each time that their frequencies and those of the parts or whole of the receiving system forming the exhaust or intake come into coincidence. This reasoning is valid because noise analyses show that the effects of coupling and interactions between these different parts can be disregarded in most cases.

The excitation frequencies can easily be exhibited clearly in the case of a piston engine. Let n_c be the number of cylinders and q the number of strokes of the cycle of the engine rotating at N rpm. The fundamental excitation frequency, equal to the opening frequency of the valves is $f_1 = n_c/q \cdot N/30$. It follows that, in a general way, in the case of the exhaust of a 4-cylinder, 4-stroke engine, for example, the excitation harmonics:

$$f_1 = \frac{N}{30},\ f_2 = \frac{N}{15},\ \ldots,\ f_n = \frac{nN}{30}.$$

f_1 is furthermore equal to h_2, harmonic 2 of the number of revolutions per second of the engine.

The existence of these harmonics is sufficiently confirmed by experience. The same list of harmonics of excitation is valid for intake noise (Part 3).

We shall confine ourselves to a single example of noise spectrum, relating to a simple system: the exhaust of the Peugeot 203. It corresponds to the most simple acoustic diagram possible since it includes a simple pipe furnished at its free end with a free passage absorption muffler. The developed length of this exhaust is (on the average for 4 valves) 4.85 m. The absorption obtained by the muffler in question being far from complete, notably in the case of low and intermediate frequencies (from 30 to 200 and from 200 to 600 Hz), the system is to be considered as a closed pipe on the valve side and open pipe on the outlet side, and can be the location of pipe resonance phenomena (standing waves).

The study of means of attenuation which follows was preceded by systematic measurements (levels and analyses of noises) intended to clearly exhibit the characteristics of this type of exhaust. The latter has great simplicity but, in return, all its disadvantages. Our goal in this experimental study carried /12 out on this type of exhaust was to improve it decisively. Testing was carried out with no load.

The low load recordings have the advantage of removing, to a very great extent, the effects of the engine noise in its true sense. Furthermore, with low load, the microphone can be installed at a very short distance from the end of the pipe.

Many noise analyses made under these conditions have allowed, together with recordings of levels corresponding to the different modes of operations studied, setting up a list of the principle components of the noise.

No matter what may be the noise spectrum considered, i.e., no matter what may be the mode of operation, it is ascertained that there is amplification of excitation components when these latter perceptibly coincide with the various resonance frequencies of the system. As it should, this amplification is furthermore increasingly pronounced as this coincidence becomes more perfect and as the frequency in question becomes lower in the case of a given operating mode.

It is observed:

1. That the low-pitched sounds are by far the most energetic components;

2. That the intermediate frequencies are always widely represented;

3. That the high-pitched sounds are very common and have much less energy.

The conclusion can be:

1. That, in order to lower the sound level corresponding to such spectra, the problem of low-pitched sounds should be attacked first of all;

2. That a rather extensive action on the intermediate frequency compo-nents will have to be contemplated;

3. That the high-pitched sounds, as well, present an attenuation problem in spite of their lack of energy.

The same conclusions can, furthermore, be extended to include the case of intake noise (Part 3).

Comments

1. The fact that any exhaust pipe of a piston engine is to be considered as closed straight above the exhaust valve, has already been stated, chiefly by Martin and Muller. This last specialist, from the Buick Motor Division, confirmed the perfect agreement existing between the theoretical distribution of nodes and antinodes of pressure and speed special to resonance modes as well as their actual distribution experimentally determined.

It will be seen that the same is true in the case of the intake of piston engines (Parts 3 and 4).

2. The problems of flow and losses of load will only be discussed in Parts 2, 3 and 4 where specific examples will be given.

CHAPTER II
MEANS OF ATTENUATION

Chief Means of Attenuation

A rational solution of the problem of noise attenuation in pipes requires /13
precise knowledge:

1. Of the frequency domain of the noise to be damped;

2. Of the value of damping required in each case.

It is known that the soundproofing of pipes becomes more difficult as the ratio of their transverse dimensions (diameters) to the wavelengths involved becomes greater. Thus, an arbitrary selection cannot be made of the values of their sections which are, in some, to be considered as given. The resources available are based on very different principles, and attenuation devices, or mufflers, to be installed on the pipes for soundproofing cannot be installed at any random location on the latter.

The conditions required for establishing an ideal soundproofing practically never occur at the same time. In most cases, the engineer's task consists in determining the one or more best solutions since the possible location of the one or more mufflers is to be determined with regard to the various data (available space, theoretical possibilities, etc.).

However, after what has been said on the subject of the simple example of exhaust with single absorption muffler located at the rear end of the pipe, it is easy to understand that the position of the mufflers will determine to a very great extent the possible modes of resonance of the various parts of the pipe resulting from the installation used.

Also, in their presence of a volume installed in series on a given pipe will have as a result a total attenuation effect on the levels and spectra in the direction of a reduction of the components at lower frequencies, taking into account the general aspect of said spectra (predominance of low frequencies).

This total attenuation, owing to the fact that the frequencies of excitation amplified according to the mechanism described are shifted toward the less energetic high frequencies in the applications concerning us, it is extremely difficult to calculate. We shall return to the subject later and will only calculate the attenuations characteristic of the different principles.

Attenuation in Its True Sense

a. Attenuation by Absorption

The absorption or "Burgess" muffler is a low-pass muffler in which, above /14
all, the high frequency components are attenuated by the phenomenon of friction of a part of the fluid in a packing of absorbent material arranged around the perforated pipe (Figure 2). Since the fluid is the location of high frequency waves, the latter are found to be damped in a more or less efficient manner. This action is exerted on those waves of velocity and correlatively on those of pressure which are associated with them. An absorption low-pass will therefore always have to be installed as close as possible to the free section of the pipe, i.e., at the rear end if it concerns a vehicle exhaust pipe. In the case of λ wavelengths which do not exceed the double thickness of the absorbing collar and, in the case of a perforated pipe with circular cross section, the attenuation produced is, according to Cremer:

$$\Delta D_{\lambda > r} \cong 3a\frac{l}{r}\ \mathrm{dB},$$

l being the length of the collar and a the absorption coefficient of the material. The perforation of the tube which allows passage of the gaseous flow has a very great effect on the behavior of such a filter. It should be staggered so as to affect all of the generating lines of the tube whether circular or not in section.

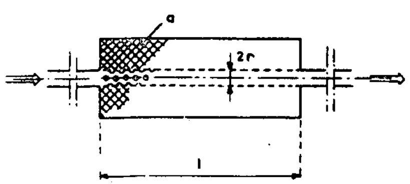

Figure 2.

b. Attenuation by Reflection

When a combination is made with one or several acoustic resistances with a piping system, the sound which is propagated there is found to be partially reflected leading, in this way, to a damping of the sound at the system outlet. This reflection can be produced in the following manners.

1. Volume in Series

Let (Figure 3) a cylindrical volume be installed in series on the pipe.

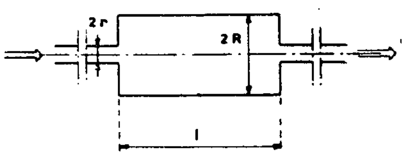

Figure 3.

There results from this installation a widening $m = R^2/r^2$ of the section. A strict calculation in the absence of reflection at the extreme end of the pipe (undefined pipe, absence of resonance) provides for the effective attenuation by reflection in the widened section produced (Bentele): /15

$$\Delta D_i = 10 \log \left[1 + \frac{1}{4}\left(m + \frac{1}{m}\right)^2 \sin^2 \frac{\pi f_i}{2 f_0} \right] \text{ dB},$$

where $f_0 = c/4l$ is the fundamental pipe resonance frequency of the widening (in quarter wave), and f_i the frequency of component D_i under consideration. The optimal damping for a given widening is achieved for:

$$\sin \frac{\pi f_i}{2 f_0} = \sin \frac{\pi 4 l f_i}{2c} = \sin \frac{\pi 4 l}{2 \lambda_i} = \pm 1,$$

i.e., for the fundamental frequency $c/4l$ and its off multiples, and its value is only a function of m.

In the case of several widenings installed in series one behind the other on a same pipe, it is only possible to be satisfied with overlapping the damping curves when they are rather far removed and not affecting each other. For example, the pipe with length L connecting two capacities (Figure 4) should be considered as open at its two ends and can be the location of half-wave resonance phenomena at frequencies:

$$f_1 = \frac{c}{2L},\ f_2 = \frac{c}{L},\ \ldots,\ f_n = \frac{nc}{2L}.$$

It is necessary for the lowest resonance harmonics of L to be located below the damping domain of the widenings reduced if it is hoped to avoid any perceptible reduction of the ΔD_i damping of the latter.

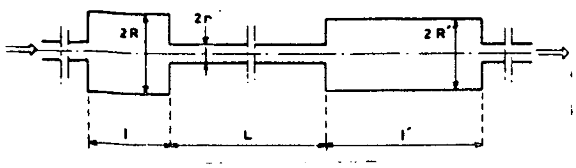

Figure 4.

Moreover, each widening (area) forms with the connecting pipe (mass) a Helmholtz resonator installed in series in the piping, hence a reduction of damping at the characteristic (single) frequency of the resonator. It is in practice easy to keep the latter rather low since it is equal, in the case of widening m and throat L, to: /16

$$f(m.l,r,L) = \frac{c}{2\pi\sqrt{ml(L + \pi/2r)}} .$$

By consolidating the various widenings of the section, there is produced a filter in series to which the preceding report is no longer applicable. The low-pass and high-pass filters, used as bypass, have definite advantages with respect to those installed in series. The abrupt widenings and narrowings, produced in volumes in series, furthermore cause losses of load which can hardly be disregarded in our applications.

2. Capacities in Bypass

The volumes installed in a bypass situation for Helmholtz resonators with or without throats (Figure 5) provide, in the case of a pipe without reflection or in the absence of pipe resonance, an attenuation:

$$\Delta D_i = 10 \log \left[1 + \left(\frac{\frac{\sqrt{c_0 V}}{2S}}{\frac{f_i}{f_0} - \frac{f_0}{f_i}} \right)^2 \right] \text{dB},$$

c_0 and f_0 being the conductivity in the characteristic frequency of the resonator:

$$c_0 = \frac{\pi r^2}{l + \pi/2r} ,$$

$$f_0 = \frac{c}{2\pi}\sqrt{\frac{c_0}{V}} .$$

Davis, Stokes, Moore and Stevens of the National Advisory Committee for Aeronautics, have studied rather intensively the theoretical and practical attenuations that can be obtained with systems with volumes in series and in bypass on pipes and showed that the experimental results agree closely with theoretical predictions.

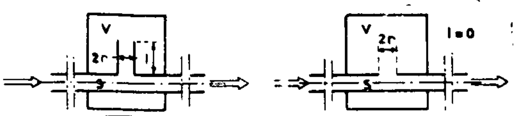

Figure 5.

3. Quarter-Wave Column Resonator

This is a pipe connected with by- /17
pass on the pipe that is to be soundproofed. The fundamental frequency of this quarter-wave resonator (Figure 6) is:

$$f_0 = \frac{c}{4l_2}$$

and its attenuation per component:

$$\Delta D_i = 10 \log \left[1 + \frac{1}{4}\left(\frac{m_0}{\frac{2\pi f_i S_2}{c_0 c} - \cot \frac{2\pi f_i l_2}{c}} \right)^2 \right]$$

where $m_0 = S_2/S_1$. When, in the expression of conductivity c_0, the correction per free end $\pi/4r_2$ is disregarded, i.e., if it is stated:

$$c_0 = \frac{\pi r_2^2}{l_2 + \pi/4r_2} \cong \frac{S_2}{l_2}$$

the attenuation becomes:

$$\Delta D_i = 10 \log \left[1 + \frac{1}{4}\left(\frac{m_0}{\frac{\pi f_i}{2f_0} - \cot \frac{\pi f_i}{2f_0}} \right)^2 \right] \text{dB}.$$

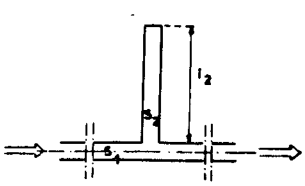

Figure 6.

Contrary to the volume resonator, the column resonator is characterized by a law of periodic function attenuation of the frequency. It is, from this viewpoint, to be associated with the abrupt widening (capacity in series on the pipe). Since it requires a clearly smaller section than the latter and since the devices which are longer, but smaller in diameter, are much easier to install on an automobile than those which (by virtue of the Helmholtz resonator principle) are very bulky, it represents one of the most valuable systems which we have at our disposal for the attenuations of low frequency components.

Comments

This results from the very principle of volume and column resonators that, when it is a matter of acting on a stationary wave, the connection of the resonator should be planned for straight above a pressure antinode of this wave.

c. Attenuation by Interference

Its principle consists in producing a division of the wave or waves so as not to reintegrate the one or more fractions branched off the main duct except at the proper moment as a function of the desired effect. /18

Let a section of pipe AD (Figure 7) on which, from D to C, the length l_1 is duplicated by a second duct l_2 with a length greater than l_1.

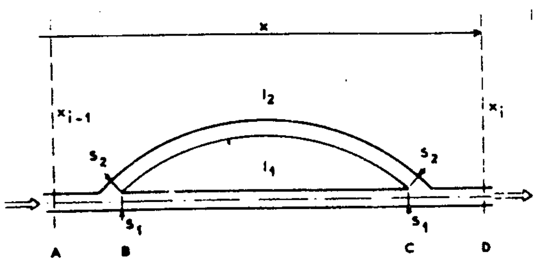

Figure 7.

Let there be in x_{i-1} the incident wave p_{i-1} (ω) of amplitude $P_{i-1,\lambda}$. The slight effect of attenuation resulting from the widening of the duct at B in the case of Figure 7 will be disregarded. In B there is a division of the wave under consideration and it is possible to hypothesize that the amplitudes of the λ wave fractions, penetrating into l_1 through S_1 and into l_2 through S_2, are affected by coefficients of distribution equal to the fractions of the total section $S_1 + S_2$ in B and in C borrowed respectively, allowing the pressures to be written:

at A:

$$p_{i-1}(\omega) = P_{i-1,\lambda} \sin \omega t;$$

at D:

$$p_i(\omega) = (1 - m)P_{i-1,\lambda} \sin \omega t + mP_{i-1,\lambda} \sin (\omega t - \phi)$$

with:

$$m = \frac{S_2}{S_1 + S_2}$$

and:

$$\phi = \frac{4\pi l f}{c}$$

if it is stated for the difference of operation employed $l_2 - l_1 = 2l$. m will be called the "opening coefficient" or merely "opening" of the system. This concept is to be consolidated with the one for "conductivity" in the case of resonators.

There is again at D: /19

$$p_i\ (\omega) = P_{i-1,\lambda}\sqrt{1 - 2(m - m^2)\left(1 - \cos\frac{4\pi l f}{c}\right)}\ \sin\ (\omega t - \phi')$$

with:

$$\phi' = \text{arc tan}\ \frac{m \sin\phi}{1 - m(1 - \cos\phi)}\ ,$$

ϕ' being the phase shift of the resultant $P_{i\lambda}$ with respect to $P_{i-1,\lambda}$.

It therefore follows:

$$\left(\frac{P_{i\lambda}}{P_{i-1,\lambda}}\right)^2 = 1 - 2(m - m^2)\left(1 - \cos\frac{4\pi l f}{c}\right)$$

and the attenuation produced is, in decibels:

$$\Delta D = 10\ \log\ \frac{1}{1 - 2(m - m^2)\left(1 - \cos\frac{4\pi l f}{c}\right)}\ .$$

The expression of attenuation by interference found above is to be reconciled with the one for the column resonator. It is very appreciably equivalent to it. If the difference in operation $2l$ (Figure 7) is accomplished by reflection of the wave fraction branched off at the closed end of a pipe installed as a bypass, there is used the principle of Figure 6, i.e., the column resonator whose attenuation becomes, in decibels:

$$\Delta D_i = 10\ \log\ \frac{1}{1 - 2(m - m^2)(1 - \cos\ \pi f_i/f_0)}$$

in which $f_0 = c/4l$, characteristic frequency of pipe l_2 closed at one end. The attenuation is maximum for the uneven harmonics of the resonance fundamental frequency of the pipe. Likewise, it is maximum in the case of Figure 7 for:

$$2(l_2 - l_1) = 4l = n\lambda \quad (n = 1, 3, 5, \ldots)$$

and its value is only a function of the opening m. In the case of difference in operation of $2l$, the cut-off frequencies will therefore be $c/4l$, $3c/4l$, ... ($f_i/f_0 = 1, 3, 5, \ldots$). The nomogram of Plate 2 is very handy to use and provides a specific idea of the above law of attenuation.

Comments

The most easily produced attenuation by interference is the one where it is produced with a quarter-wave column resonator. In reality, the one described above and corresponding to Figure 7 (paragraph c) can only be used in this form on an exceptional basis.

The effective production of differences in operation sought for is difficult and cannot in any case be accomodated to empirical devices. This fact has been emphasized quite recently by W. Buerck. It will be seen that, although interference itself cannot cope with soundproofing problems which arise, it does represent in certain specific cases an indispensible resource.

Total Attenuation

The preceding already provides an idea of the general structure of mufflers. It is clear that the total attenuation defined above will be a function of the total length of the pipe considered, the number and distribution of mufflers on this pipe and their design. Like attenuation itself, it will likewise be a function of the operating mode, i.e., excitation frequencies of the system under consideration. It comes from phenomena of reflection in the pipe, a reflection which gives rise, for certain operating modes, to the establishment of stationary waves in the sections of pipe determined by the one or more intermediate mufflers. These sections are shorter than the whole pipe itself and, if these sections of pipe are counted beginning from the valves (exhaust and intake), the first one will have resonance modes in quarter-waves and the following ones resonance modes in half-waves. /20

Each time that the question of total attenuation arises, it will concern intermediate mufflers, the terminal muffler or the free end of any pipe always giving rise, in a more or less accentuated manner, to reflection.

Consequently, it is possible to state that any intermediate (nonterminal) muffler will attenuate, by the fact alone of its presence, the low-pitch components since the modes of pipe resonance are found to be shifted towards higher frequencies.

This attenuation will not be calculated since we believe it is practically impossible. Nevertheless, it will be possible to apply a qualitative judgement on the phenomenon which allows above all carrying out a graphic method for determining low frequencies for high-pass calculation (Part 2).

Comments

When the problem of soundproofing exhaust and intake is considered in all its complexity, i.e., when, in addition to the purely acoustical aspect, the questions of losses of load and bulkiness, without which everything would be quite simple, are taken up, it is very quickly seen that many of the theoretically advantageous solutions are to be rejected.

As will be seen below, there is, in the case of the exhaust, one additional difficulty: the high value of the speed of sound which has as a result a requirement for volumes and lengths of throats, and therefore much larger dimensions for a given effect than for the case of the intake.

CHAPTER III
THE ORDERS OF MAGNITUDE

After this overview of excitation spectra and principles of characteristic attenuation which could be used for exhaust and intake, it appeared useful to provide here some details on the subject of orders of magnitude and their bearing on problems of soundproofing which occur in practice. At the risk of appearing premature, they will help in understanding what follows. /21

Exhaust (Part 2)

The exhausts are most often pipes with total lengths of from 4 to 6 m in the case of private vehicles. They are furnished with one or several mufflers and, in spite of the mean high values of speeds of sound (on the order of 500 m/s), the lengths of pipes remain on the order of the half and quarter of possible stationary wavelengths.

The problem of soundproofing the exhaust should therefore essentially take into account resonance phenomena of pipes, to the practical exclusion of volume resonances, since the characteristic frequencies of Helmholtz resonators are believed to be considerably less than the lower excitation frequencies.

Intake (Part 2)

The intake ducts are relatively short but are generally connected to closed spaces (filter-mufflers), themselves associated with throats. The orders of magnitude of the sizes of these ducts and these throats are not systematically those of half and quarter wavelengths of principal components of the excitation spectrum of the intake. On the other hand, associated with the capacities mentioned, these throats form Helmholtz resonators whose characteristic frequencies are always on the order of magnitude of the frequencies of the main components of the spectrum.

The problem of soundproofing the intake should therefore take into account chiefly resonance phenomena of volumes.

Filling In (Part 4)

Although the resonances of pipes generally appear as secondary ones in the characteristic soundproofing plan for the intake of high-speed system engines, the same is not true for acoustic supercharging. /22

It will be seen that, along this line of thought, an attempt is made, on the other hand, to cause them judiciously in the intake duct connecting the valves to the filter housing, while at the same time taking the desired precautions in order to avoid their becoming apparent in the form of noise.

PART 2

SOUNDPROOFING OF THE EXHAUST

CHAPTER I

SPECIAL PROBLEMS WITH THE EXHAUST

The goal of Part 2 is to show what systems can be devised to cope with the problem of soundproofing the exhaust. In order to solve this problem, essentially dominated by very restrictive installation conditions, we have studied the possibilities offered by the various principles of attenuation compatible with the appropriate data. /25

Origin and Effects of Exhaust Noise

Contrary to a gas turbine with the same power, the piston engine involves a relatively low output, but, on the other hand, a very high pressure. Since expansion issprevented, this pressure is not entirely transformed into work and the pressure still prevailing in the cylinders at the instant the exhaust valves open amounts to several atmospheres. On the other hand, the flow is pulsed and the beginning of the exhaust puffs occurs ~~out~~ at very high velocities at the valves. Another essential fact is that this flow takes place in the exhaust pipe whose acoustic behavior is rather complex.

As far as the passengers of the vehicle are concerned, the noise from ordinarily soundproofed exhaust appears slight, although the other noise components are great. On the other hand, when these latter components are highly attenuated, the exhaust must itself be very well soundproofed. The "exhaust" component increases by several decibels the level of noise in a vehicle such as the Peugeot 203 with windows closed. In order to calculate the effect of the exhaust noise alongside the road we carried out measurements at halt (engine idling) at various points distributed around the vehicle (Plate 3), microphone at 1.70 m from the ground. The number pairs plotted directly above the measurement points provide the values in decibels of the engine and exhaust components for a Peugeot 203, successively equipped with an exhaust in series and an exhaust improved by attenuation of the highest energy components, as will be seen in the following. The central figure provides the ratios of engine/exhaust sound energies according to the explanations at the bottom of Plate 3. It will be noted that the differences, revealed by these measurements, are more pronounced in load, i.e., in normal operation of the vehicle.

In conclusion, it can be said that from the point of view alone of people alongside the road, effective soundproofing of the exhaust is a necessity.

Program Followed

It has been seen that, in a general way, the low-pitch components predominate in any exhaust noise, chiefly in the one produced with an absorption muffler at the rear end of the pipe. The general tendency to lower body floors brought on by recent developments in body design has led to consideration as a basic hypothesis the obligation to house the muffler in the vicinity of the extreme end of the pipe, provided there is one alone, or one of the mufflers at this place, in the case of versions with several mufflers. /26

Since the absorption muffler acts very well on high-pitched sounds, it appeared advantageous for us to examine an exhaust, first of all, the capabilities of joining the absorption low-pass and the high-pass involving one of the principles of characteristic attenuation described or several of them simultaneously.

In the various possible versions and relating to one given goal, the different attenuation effects should compliment one another. In this way, when the total attenuation of an intermediate muffler is very pronounced, it is possible to be satisfied with an attenuation characteristic of mufflers which are weaker in the band of dominant (LF) frequencies than in the case where the above-mentioned effect is weak.

Furthermore, when the total attenuation is sufficient, the attenuation characteristic of the intermediate muffler can be pronounced in the HF band. In consequence, it is possible in this case to reduce the attenuation characteristic of the terminal muffler in the HF band at the expense of an accentuation of its characteristic attenuation in the IF band.

Having passed this stage, it will be shown that it is possible to design an exhaust with a single (terminal) muffler without an absorbent (using the principle of reflection, interference or friction) with a very low loss of load and which does not have the conventional defect of the terminal muffler without absorbent which will be taken into account.

CHAPTER II

COUPLING OF THE HIGH-PASS AND THE LOW-PASS

Low-Pass

Taking into account the high efficiency of the absorption low-pass, this study was carried out making use of this type of terminal muffler. It has already been stated that even for this type of essentially simple muffler some conditions must be complied with. In general, these free passage mufflers are well designed. Survais (England) made efforts to design a substantial improvement in the service life of the absorbent which, without precautions, rapidly disintegrated under the effect of variations of pressure owing to the pulsing character of piston engine flow as well as vibrations and condensations. /27

The best absorbent is fiberglas with long fibers wound in a coil. The optimum volumetric weight is located in the vicinity of 105 kg/m^3, but the efficiency of the muffler only varies slightly as long as this volumetric weight is kept between 90 and 120 kg/m^3. When the porosity of the absorbent exceeds a certain boundary, the absorption muffler is transformed into a reflection muffler with damping by friction whose characteristic attenuation is extended to a frequency band which becomes narrower when the volumetric weight of the absorbent becomes smaller.

The undeniable advantage of the absorption muffler is its great simplicity. However, its effect is practically zero in the L. F. band (30-300 Hz) in the case of acceptable dimensions for an automobile vehicle. It will also be noted that the noise produced with a single absorption muffler, installed in the rear as it should be, is a muffled sound, with few high-pitched but many low-pitched sounds and very intense as will be seen by comparison with other systems which will be studied.

Optimum Position of Any Nonterminal Muffler

Let there be a muffler S_1, nonterminal, forming part of an exhaust (Figure 8): L_1 represents the mean distance developed (from the exhaust valves s to

muffler S_1) from the first part of the pipe and L_2 that of the second part of the pipe (from muffler S_1 to the muffler S_2). The total attenuation of /28
muffler S_1 becomes better as the condition of nonsimultaneity of resonances of L_1 (pipe closed) and L_2 (open pipe) is better confirmed. It should follow that:

$$\frac{c(T_1)}{4L_1}, \frac{3c(T_1)}{4L_1}, \frac{5c(T_1)}{4L_1}, \dots \neq \frac{c(T_2)}{2L_2}, \frac{c(T_2)}{L_2}, \frac{3c(T_2)}{2L_2}, \dots.$$

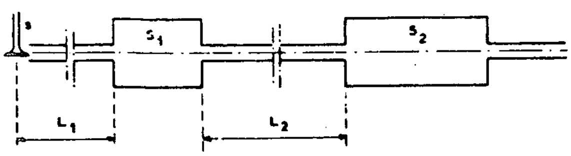

Figure 8.

Taking into account the orders of magnitude, it is advantageous to have, whenever there is the possibility:

$$L_2 = 3\sqrt{\frac{T_2}{T_1}}\, L_1.$$

T_1 and T_2 being the mean temperatures of the gases in L_1 and L_2. The temperature of the exhaust gas varies, on one hand, with the load and on the other hand, with the operating mode of the motor at a given point. Furthermore, heating the pipe by contact and cooling down by convection and radiation, the gases have a temperature which decreases with distance, from the measuring point, of the exhaust valves (Plate 4). Since it is under load conditions that the levels of noise at different operating modes are the greatest, the temperatures in load are therefore to be taken into consideration. The curves of Plate 4 are the result of recordings on a vehicle under full load for an ambient temperature of 20°C in the shade and can be considered as valid for the whole following study.

Determination of the Frequencies for Calculation of High-Passes

The diagramatic method illustrated by Plate 5 allows a very quick determination of the low critical frequencies of a two-muffler exhaust.

Since the total developed length of the exhaust is prescribed, L_1 and L_2 are provided taking into account the maximum predicted length of the high-pass.

A diagramatic determination is then made of the excitation fundamentals of the engine corresponding to the operating modes giving rise to coincidences of harmonics of excitation and harmonics of resonance of the most harmful values of L_1 and L_2.

The critical frequencies will only be definitive when the length of the calculated high-pass is consistent with predictions. These predictions are based, just like the concept of the high-pass, on a wise selection of the /29 attenuation principle suitable for use.

When they are found to be false, a start is again made taking into account the orders of magnitude found. Two preliminary projects generally are enough.

The Column (or Reflection) High-Pass

It is easy to gain confirmation that the column resonator represents the least bulky high-pass. It lends itself quite well to construction and length calling upon the principle of concentric columns of annular sections interconnected at their ends. In this way, resonance pipes are produced as quarter-waves folded back several times on themselves. The high-pass represented schematically in Plate 6 includes three successive high-passes calculated for three critical (fundamental) frequencies:

$$f_1 = \frac{c(T_1')}{4(3l_1' - l_0)} = 106 \text{ Hz},$$

$$f_2 = \frac{c(T_2')}{4(3l_2' - l_0)} = 83 \text{ Hz},$$

$$f_3 = \frac{c(T_3')}{4(3l_3' - l_0)} = 120 \text{ Hz}.$$

The calculation temperatures T_1', T_2' and T_3' of three quarter-wave columns are mean temperatures under load, measured in the same conditions as those which refer to the same exhaust without high-pass (Plate 6).

The lengths l_1'' and l_2'' (Plate 8) represent the distances from the pressure tap of the first column to that of the second and from the pressure tap of the second column to that of the third, and, for them too, conditions of nonsimultaneousness of resonances are to be observed. When c_1'' and c_2'' are the respective velocities of sound in l_1'' and l_2'', it must follow that:

$$\frac{c_1''}{2l_1''} = nf_1,\ n'f_2,$$

$$\frac{c_2''}{2l_2''} = n''f_2,\ n'''f_3.$$

in which n, n', n'', n''' $\neq$ 1, 2, 3, ... In the case of our high-pass, it follows that:

$c_1''/2l_1''$	f_1	f_2	n	n'
717	106	83	5.9	7.6
$c_2''/2l_2''$	f_2	f_3	n''	n'''
500	83	120	6.2	4.3

It will also be noted that the outside diameter of this high-pass is remarkably low (is equivalent to 2d, d being, at approximately the thickness of the plate, the diameter of the exhaust pipe) in spite of the fact that the optimum value of the ratio of the section of that of the pipe, $S_2/S_1 = 1$ (cf. Figure 6), was accepted, for the three basis resonators./ /30

Results

The attenuation produced by connection of this triple high-pass is remarkable. Plate 7 only provides extracts of the recordings of spectra and levels carried out with this exhaust under no load, microphone at 25 cm from the outlet pipe and at 45° from the mean direction of flow. In no case was attenuation of the level less than 13 dB. It is 18 dB at 2,500 rpm and varies between 17 and 25 dB under load. Recall that a gain of 15 dB corresponds to a reduction of sound energy by a little more than 96%.

Mixed High-Pass (With Reflection and With Interference)

In spite of the small radial dimensions of such a column high-pass (its outside diameter does not exceed 66 mm), the length resulting from the juxtaposition of three elementary high-passes for calculation temperatures on the order of 650°K can be excessive. It appeared advantageous for us to try to reduce this. In order to preserve the system efficiency, it is a matter of widening the attenuation bands (cf. expression of characteristic attenuation of the column resonator) of the quarter-wave column for the purpose of only preserving one of the three. This problem is, in sum, the same thing as increasing the number of cut-off frequencies of a column and spreading out the attenuation characteristic of the resultant system in the IF and HF bands.

Plate 8 shows such a high-pass. It can be characterized as mixed, not because it operates on two principles of attenuation in its true sense (reflection in the case of LF and interference in the case of the IF and the HF high-passes), but because it likewise possesses low-pass characteristics.

The increase in the number of cut-off frequencies is produced by a string of perforations contrived in the casing corresponding to the first reversal of the LF column. This connecting of the first two parts of this column, located at a distance l_2' from the first reversal (rear end of the muffler, cf. Plate 10) has the effect of causing the appearance of a second LF column overlapping the first one (which is found to be partially short-circuited). It will be seen a little further on that the effect of this multiplication (by 2) of the number of cut-off frequencies is clearly beneficial.

As far as the IF and HF components of the spectrum, it is found that a part of the quarter-wave column high-pass (whose length remains great even when the columns are folded back twice on themselves) lends itself very well to the production of advantageous differences in operating mode.

In spite of the relatively low value of the reduction of total level that can be produced by attenuation of the IF and HF components, taking into account the preponderance of low-pitched sounds, it is easy to confirm — as much for the level as for the tonal quality — that this capability for damping the equipment as well as high-pitched sounds merits its further use: /31

a. Capabilities for Reduction of Level by Attenuation of IF and HF Components

Let there be a noise spectrum of exhaust with components

$$D_0 \geq D_1, D_2, D_3, \ldots, D_n.$$

All the spectra show the preponderance of low frequencies. These latter will be designated by D_0. The interference system to be determined is intended to act on the IF and HF, i.e., on $D_1, D_2, \ldots, D_n$. With a difference of amplitude $D_0 - D_i$ in decibels, corresponds a ratio of energies ρ_i; and there is for a total level at a given mode of operation, and for the exhaust with single terminal low-pass to be improved,

$$D = D_0 + 10 \log \left(1 + \frac{1}{\rho_1} + \frac{1}{\rho_2} + \ldots + \frac{1}{\rho_n}\right).$$

If the high-pass is connected to this, it follows that:

$$D' = D_0' + 10 \log \left(1 + \frac{1}{\rho_1'} + \frac{1}{\rho_2'} + \ldots + \frac{1}{\rho_n'}\right).$$

It is thought to make maximum the difference D - D', which amounts to the same thing as acting optimally low frequencies in order to produce the attenuation $D_0 - D_0'$ which is as substantial as possible and to make minimum the quantity

$$\Delta D' = 10 \log \left(1 + \frac{1}{\rho_1'} + \frac{1}{\rho_2'} + \ldots + \frac{1}{\rho_n'}\right).$$

An examination will be made of a method for attenuation of the IF and HF components by simple interference which consists in acting energetically on a single well-selected calculation frequency with an optimum aperture coefficient m'.

The optimum value m' of the aperture coefficient is provided by:

$$\frac{d}{dm}(m - m^2) = 0$$

and is equal to 0.5. Let us consider the sum of $D_0', D_1', \ldots, D_n'$.

With respect to D_0' already attenuated by reflection, the quantity 10 log $(1 + n/\rho')$ represents an increase which must be minimized by action on a well--selected frequency f_m.

If there is assumed for the I. F. and H. F. part of the spectrum:

$$\sum_1^n D_i' = D_1' + 10 \log n.$$

and it has been seen that it is possible, by interference, to correct this /32
sum to:

$$\sum_1^n D_i'' = D_1' - 10 \log \frac{1}{1 - 2(m' - m'^2)\left(1 - \cos \pi \frac{f_1}{f_m}\right)} +$$

$$+ 10 \log \left(2 - \frac{1}{\rho_2''} + \frac{1}{\rho_3''} + \dots + \frac{1}{\rho_{n-1}''}\right),$$

when f_m is the mean frequency of the IF and HF part of the spectrum.

In order to determine ρ_i'' it will be noted that:

$$D_1' - D_i'' = 10 \log \frac{1}{1 - 2(m' - m'^2)\left(1 - \cos \pi \frac{f_i}{f_m}\right)}$$

and that:

$$D_1' - D_1'' = 10 \log \frac{1}{1 - 2(m' - m'^2)\left(1 - \cos \pi \frac{f_1}{f_m}\right)},$$

where:

$$D_1'' - D_i'' = 10 \log \rho_i'' = 10 \log \frac{1 - 2(m' - m'^2)\left(1 - \cos \pi \frac{f_1}{f_m}\right)}{1 - 2(m' - m'^2)\left(1 - \cos \pi \frac{f_i}{f_m}\right)}$$

and that the action on a single mean frequency f_m will provide an attenuation $\sum_1^n D_i' - \sum_1^n D_i'$ of the IF and HF part of the spectrum in decibels:

$$G_1 - 10 \log n - 10 \log \frac{1}{1 - 2(m' - m'^2)\left(1 - \cos \pi\frac{f_1}{f_m}\right)}$$

$$- 10 \log \left[2 - \sum_{2}^{n-1} \frac{1 - 2(m' - m'^2)\left(1 - \cos \pi\frac{f_i}{f_m}\right)}{1 - 2(m' - m'^2)\left(1 - \cos \pi\frac{f_1}{f_m}\right)} \right].$$

It should be noted that this attenuation essentially is related to the system of IF and HF bands with the almost total disregard of the LF band.

Several successive actions on as many different frequencies can only be carried out with a system with a much greater total elongation which would arise from our hypothesis. This is the reason it is wise to select as an interference device, capable of being integrated in a mixed high-pass of the type shown in Plate 8, a system intended above all to have a favorable effect on the tonal quality of the resultant noise.

b. Capabilities for Improvement of the Tonal Quality by Attenuation of Equipments and High-Pitched Sounds

The principle of attenuation by interference lends itself especially well to the suppression of high-pitched sounds when room available is reduced. It is then the total or partial division of the pressure waves which forms an excellent method. References made to the end of Part 2 where we /33 have chosen to study their principle and the various possibilities.

As shown in the diagram of Plate 8, this mixed high-pass includes an IF and HF system with a partial division of waves. There may therefore be found, in the wall of chamber A, apertures which form:

1. The pressure tap common to the 2 overlapping LF columns, defined above, with $S_2S_1 = 1$, whose fundamental cut-off frequencies graphically determined are:

$$f_1 = \frac{c(T_1')}{4(3l_1' - l_0)} = 81.5 \text{ Hz}$$

and:

$$f_2 = \frac{c(T_1' = T_2')}{4(3l_1' - 2l_2' - l_0)} = 102 \text{ Hz};$$

2. The section furnishing passage, with an impedance of $\rho c/S_2$, to the dF nad HF wave fractions, reflected and reintegrated with the main duct along mixer C. A great number of operational differences is produced in this way.

Results

This mixed high-pass is with division of flow. This is how it is stated in Plate 8 in order to differentiate it from other possible versions. Its loss of load is extremely low, although the loss to abrupt widening at the intake was unavoidable with $l_1' = 500$ mm.

Plate 9 provides the curve of the total level (a) produced with this solution of mixed high-pass, associated with the same terminal low-pass as the one giving rise to the greater noise curve when it is installed along at the rear (s). The maximum gain reaches 15 dB at 2,250 rpm in the case of $L_2/$ $/L_1 = 2.64$ (cf. Figure 8). If the pressure tap of the second LF column is removed (opening E located at $l_2' = 240$ mm from the rear end), the high-pass then has no more than one LF column and the gain in decibels decreases perceptibly beginning from 3,100 rpm.

In this case, there is lost approximately 4 dB from 3,250 rpm to 3,750 rpm and more than 5 dB at 4,000 rpm and beyond. This is logical since the fundamental cut-off frequency of the second column removed in this way was 102 Hz (corresponding to 3,060 rpm).

Conclusion

Other versions of mixed high-pass filters are possible. We do not show them because they lead to the same conclusion:

The coupling of column high-passes with absorption low-passes leads to advantageous solutions when it is a matter of producing very low outside diameters. The total attenuation of a column high-pass is slight but for a total given bulkiness of the mufflers, it is hardly possible to do better. /34

When it is impossible to build in length, the tendency is to increase the total attenuation of the first muffler by increasing diameter. Since only the principle of the column resonator allows design of high-pass filters with as low total volume, the muffler with total increased attenuation cannot be a high-pass. It will be — according to the hypothesis stated at the beginning of this part — a low-pass or a mixed low-pass (acting also in the intermediate frequencies).

CHAPTER III

COUPLING OF TWO REFLECTION MUFFLERS

Principle

When the first muffler is a mixed low-pass, the terminal muffler can also /35
be a mixed filter, designed preferentially for a suitable characteristic attenuation in the IF band. This will therefore be a mixed IF filter (also acting in the HF and LF bands).

The attenuations:

1. Total from the first muffler (mixed low-pass);

2. HF and IF from the first muffler;

3. HF, IF and LF from the second muffler (main and terminal),

have in this way the greatest opportunity to mutually supplement one another at all operating modes.

HF and IF Mixed Filters

It has been seen (Part 1) that the volume resonator only has a single cut-off frequency beyond which the characteristic attenuation decreases rather swiftly. Any perforated tube passing through a volume which cannot be assimilated to one column forms a volume or Helmholtz bypass resonator. In order to widen the band of frequencies involved by these simple systems it is possible — within the available volume of the muffler — to provide for several, each one having a different cut-off frequency. Such a multiple system (Figure 9a) is always more advantageous than a system with a single resonator. Indeed, it can be shown that it is more advantageous to moderately attenuate several components (corresponding to several calculated frequencies) than attenuating energetically a single one of them (corresponding to a single cut-off frequency).

It corresponds to the electrical diagram of Figure 9b. The frequency band involved can be considerably widened if several wave passages are contrived in a same volume as shown in Figures 10a and 10b.

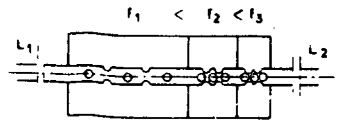

Figure 9a.

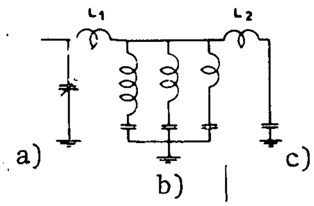

Figure 9b. a, Engine; b, muffler; c, atmosphere.

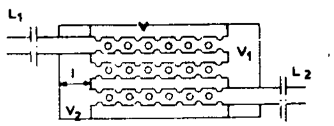

Figure 10a.

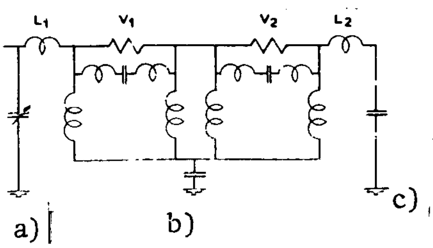

Figure 10b. a, Engine; b, 3-tube muffler; c, atmosphere.

Depending on the attenuation sought after, a three-tube muffler can be a triple-passage type in a volume V (Figure 10) or a two double--passage type in two volumes V' and V" (Figures 11a and 11b: two reflecting bends). /37

The terminal mufflers shown in Plates 10, 11 and 12 (two reflecting bends) correspond to the arrangement shown in Figure 11.

Justification for the Figures

In the case of a given resonator volume, the cut-off frequency becomes lower as the total section of perforations of the pipe which it crosses becomes smaller. Now the calculated frequency of the resonator cannot compare with the frequency in the characteristic meaning of the term unless the aforementioned section is of the same order as that of the pipe itself. The result is that the characteristic attenuation of the system will extend that much better into the low frequencies as the volume becomes longer for a determined section.

The connecting chambers at the end of parallel pipes, V_1 and V_2 (Figures 10 and 11) play the role of resistances in the analog electrical circuit (characteristic attenuation by reflection from section widenings). In the case of a given total length of muffler, the relative advantage of their characteristic attenuation cannot, nevertheless, justify a lengthening ℓ (Figure 10a) greater than 0.40 D, D being the inside diameter of the assumed cylindrical revolving muffler. The losses of load corresponding to the two reversals of direction of flow, taking place in these chambers, no longer practically decrease beyond values of l given by $0.30 \leq l/D \leq 0.40$, when the total section of the perforations of each of the pipes is equal or greater than their straight section (beginning section). Being given that the losses in passage of gases /38 through V_1 and V_2 (abrupt widening and narrowing, change of direction) are by far the most important of all those which take place in such a muffler, their sum being, under these conditions, reduced to the quarter of that which it would be in the absence of perforations.

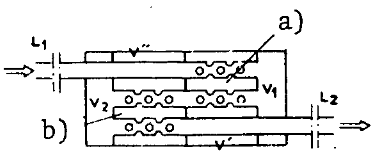

Figure 11a. a, First reflecting bend; b, second reflecting bend.

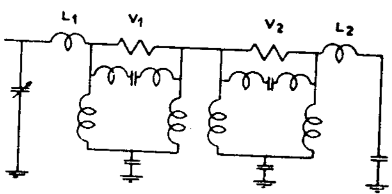

Figure 11b.

It is therefore possible to design multiple-passage mufflers with very low losses of load.

As for total attenuation, it becomes more marked:

1. As the total volume increases;

2. As the total section of the perforations becomes larger.

When for reasons of bulk, the first condition can only be really satisfied by a primary muffler, the second muffler, on the other hand, should always perform this function.

It is therefore possible to state, in conclusion, that as far as the perforations are concerned, there is no incompatability between the acoustic conditions (characteristic and total attenuations) and dynamic conditions (losses of load) to be complied with.

Type of Systems

The characteristic attenuations of the two mufflers and the total attenuation of the primary muffler should be complimentary.

Let us consider, for example, the exhaust which we designed for the Peugeot 403. In this exhaust, the small diameter (100 mm) and the small length (180 mm) of the primary low-pass, of the same type as that shown in Plate 13, as well as the impossibility of producing an optimum ratio L_2/L_1 (cf. Figure 8) forms part of the data of the problem.

Referring to Plate 10 (terminal muffler of this system), it can be readily seen that care was taken to perform an energy attenuation in the IF band (two reflection bends) intended to make up for the lack of total attenuation owing to the small dimensions of the primary low-pass.

The recordings of Plate 14 allow a precise idea to be gained of the total attenuation obtained with this low-pass which was connected for this purpose to a terminal absorption muffler. It is low.

The recordings of Plate 15 show first of all the clear superiority of the reflection muffler of Plate 10 to the absorption low-pass.

As for the connection of the mufflers of Plates 13 and 10 (403 exhaust in series) Plates 14 and 16 illustrate its advantages in spite of the low dimensions of the primary low-pass.

When the primary muffler sufficiently involves the band of intermediate frequencies at the same time preserving its effect on the high-pitched sounds, the characteristic attenuation of the main muffler can be reduced in the aforementioned band at the expense of an extension to lower frequencies. The output stage, more modest in length, can be reserved for a less widespread high frequency attenuation. In other terms, when the primary muffler is a sufficiently effective low-pass, the HF perforations of the terminal muf- /39 fler can be replaced by slots (not difficult in design): this is the case of the system of primary mufflers (Plate 17) and terminal mufflers (Plate 11). Although the primary muffler can be developed for the purpose of a rather significant IF attenuation and that, owing to this fact, its total attenuation is quite marked, the terminal muffler can only include two HF stages: as in the case of the system shown in Plate 13[1] and Plate 12. Finally, when the characteristic H. F. attenuation of the primary muffler is especially low (lack of space), it is necessary to make up for this defect by an additional action on the most high-pitched components of the spectrum in the main muffler: as in the case of the filter shown in Plate 18.

[1]The muffler of Plate 13 having in this case a length on the order of 300 mm and an inside diameter (circular section) of 110 mm.

The Case of a Very Short Length L_1

This especially advantageous arrangement will be studied in the chapter dealing with filling in. It is found that, in this case, the primary muffler can be reduced to its most simple expression, especially when it is installed in bypass at the end of the exhaust collector, as in the case of the Peugeot 403 exhaust shown in Plates 50 and 57. We shall only consider here the acoustic characteristics resulting from this arrangement.

The adaptation, at a point as close as possible to the exhaust valves (sources of pressure and antinodes of pressure of stationary waves), of a simple volume on the order of the cylinder capacity of the engine has the effect of reducing to a great extent the variations of pressure directly above the exhaust manifold.

Without this volume, i.e., without this free section, in front of the pipe, the latter being furnished with a single muffler behind, the lowest fundamental resonance frequency would be c/4 L. With a proper free section, it passes to c/2 L. A good total attenuation is therefore ensured.

As for the characteristic attenuation obtained, it is that of a Helmholtz HF resonator whose conductivity can be selected under better conditions than in the case of a small primary low-pass with $L_1 \neq 0$ (in series on the exhaust pipe).

Since this Helmholtz resonator is very slightly damp, the terminal muffler to be connected to this arrangement of primary volume is preferentially a three-stage filter, of which two are output HF, as shown in Plate 19.

CHAPTER IV
SINGLE MUFFLERS IN THE REAR

Difficulty of the Problem

We believe that we have clearly exhibited the fact that, in any well--soundproofed exhaust, the different attenuations should, as much as possible, compliment one another. For reasons of clearness, we have nevertheless only discussed up to this point attenuations of levels and components at constant speeds of the engine. When the necessary precautions are not taken, an extremely disagreeable phenomenon takes place with an abrupt cut-out of the flow beginning from a certain operational mode located in the vicinity of 3,800 rpm in the case of a 4-cylinder, 4-stroke engine such as the Peugeot 403 or 404, for example, If the intake valve of such an engine is abruptly closed, a very intense HF wave, heavily modulated in LF, is created by the closure of the exhaust valves, and becomes obvious externally through a series of crackling sounds which are intolerable owing to thier tonal quality and intensity. /40

But what precautions are to be taken? We can be reassured for the cases of those systems studied up to this point. The nonexistence of the cut-out noise is one of a quality criteria of any exhaust suitably studied and produced. Taking into account the very definition of total attenuation, it is clear that the most unfavorable location for any reflection muffler is to be found at the rear end of the exhaust pipe. Its total attenuation is then more reduced since it is, by definition, made up of volumes[1]. The same is not true of the absorption muffler which, owing to the acoustical friction which is produced there for the purpose of an optimum damping of the HF velocity waves, is characterized by a total attenuation which cannot be disregarded in spite of its installation in the rear (obligatory).

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[1]Which acoustically make up free sections at the end of the pipe.

The cut-out noise, spectacular with free exhaust, takes place with any exhaust with single muffler in the rear provided that the insufficiency of its total attenuation is not compensated for by a suitable characteristic HF attenuation.

In this way, the mufflers of Plates 10, 11, 12 and even 18 and 19 show this defect when they are installed alone and far to the rear of the exhaust pipe.

Although the problem of soundproofing exhaust is difficult for 4-cylinder /41
4-stroke engines owing to the fact of the orders of magnitude entering into the situation, the problem of suppression of cut-out noise at high speed was only able to be solved at the cost of still more considerable efforts. In the absence of a primary muffler, the energy of the wave train initiated at the cut-out of the flow is such that the casings of the mufflers themselves begin to vibrate.

It will be shown first of all how the cut-out noise can be analyzed and, subsequently, that it is possible to design a muffler with single terminal reflection, including an HF moderator suppressing all cut-out noise without involving a costly loss of load. We shall take up successively absorption, interference and reflection moderators.

Analysis of the Cut-Out Noise

Plate 20 shows one of the many polar oscillograms chosen for analysis of this phenomenon. By means of a microphone-decibel meter-filter-polar oscillograph chain with scanning controlled by the motor speed, it was possible to photograph the spot rotating at this speed, since the analyzer was adjusted for maximum amplitude. As a function of the description selected, there was obtained under these conditions 1, 2, 3 ... curves of variable amplitude (essentially transitional phenomenon) and of phase likewise slightly variable (the engine decelerating).

Knowing the speed of the engine with the abrupt cut-off of the flow at no load (the test was made at no load since under these conditions the motor is not harnessed to any external inertia, ~~of the brake style or other~~ from the brakes or otherwise.), a deduction is made from the photos produced of the frequency of the preponderant component. Since it concerns 4 cylinders, it can be seen that the phenomenon is initiated twice per revolution and that the wave is more or less damped between two excitations.

The frequency of the preponderant component was found to be equal to 1,330 Hz at 1,000 rpm. It therefore concerns the harmonic 10 of the closure frequency of the valves.

Moderators

They are designed to attenuate the high frequencies involved and are, consequently, low-pass in type integrated with terminal reflection mufflers according to the diagram of Figure 12.

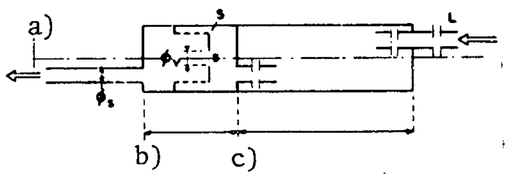


Figure 12. a, Atmosphere; b, moderator; c, main attenuation component.

Considerations of distribution of attenuations to be produced tended to reserve 200 mm in length for the moderator and 360 mm for the reflection part out of 560 mm available in all[1]. For the same reason, it can be said that the double reflection passage of the type shown in Plate 10 lends itself better for the design of the principal part of the muffler.

1. Absorption Moderator

This is a small low-pass made of fiberglas. It gives complete satisfaction /42 but does not represent in the true sense of the word a solution to the problem since it has the defect already emphasized of poor mechanical service life inherent in this kind of absorbent which, nevertheless, is acoustically the best. It suppresses the cut-out noise and provides, like the other systems using reflection and HF moderator, curves of level and intermediate noise spectra between those of a single absorption low-pass and the two reflection mufflers (Plates 14 and 16), which is logical.

[1] These figures are given by way of example. A moderate length of 200 mm is necessary and in general sufficient.

2. Interference Moderator

It will be seen below that interference, used wisely, can represent a very valuable means of attenuation in upper frequency bands although 3 dB are gained only under the best conditions for a phase shift of the fraction of amplitude deflected from $\pi/2$ (Figure 7). The total or partial division of the waves has several advantages over the simple system diagrammed in Figure 7:

It gives rise to a better distribution of attenuation;

Its attenuation only becomes 0 in the case of $l/\lambda = 0$;

It lends itself to overlapping with a direct acoustic effect of an effective irreversibility of flow;

It makes the carrying out of differences of operational mode easier.

We will show successively the total division and the partial division of waves.

a. Attenuation by Total Division of Waves

Let us consider an interference system based on the principle of Figure 7 but including t successive paths $\frac{2t}{t}$, $\frac{4t}{t}$, ..., $\frac{t-1}{t}\,2l$, $2l$ of equal impedances (Figure 13).

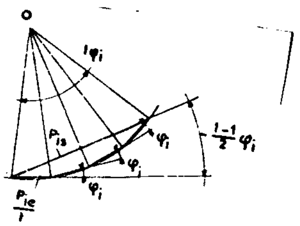


Figure 13.

To two consecutive paths correspond two fractions P_{ie}/t of the total amplitude of pressure P_{ie} at the intake, phase shifted with respect to each other by:

$$\phi_i = 2\pi \frac{2l}{t\lambda_i} = \frac{4\pi l f_i}{tc} .$$

When P_{ie} is the resultant amplitude at the output, the attenuation for /43
a component D_i of frequency f_i is:

$$\Delta D_i = 10 \log \frac{P_{ie}^2}{P_{is}^2} = 10 \log \frac{t^2 \sin^2 \phi_i/2}{\sin^2 t\phi_i/2}$$

or again, since l/t is still very small:

$$\Delta D_i = 10 \log \frac{4\pi^2 l^2 f_i^2/c^2}{\sin^2 2\pi l f_i/c} \text{ dB.}$$

The resultant phase shift of P_{is} with respect to P_{ie} is:

$$\phi_i' = - \frac{t - 1}{2} \phi_i = - \frac{t - 14\pi l f_i}{2 - tc} \cong - \frac{2\pi l f_i}{c} = - \frac{2\pi l}{\lambda_i}$$

and corresponds to a difference of operation resulting from l with respect to the incident wave P_{ie}.

Plate 21 provides the above attenuation. It is only reduced to zero in the case of $l/\lambda = 0$ and has maximum values for $l/\lambda = K/2$, with an even numbered K. Its minimum values are in logorithmic progression and correspond to $l/\lambda = (2K + 1)/4$. Our investigations were limited to the values of l/λ included between 0 and 0.75 and allowed rather reliable confirmation of the calculated values of Plate 21.

From the point of view of flow, such a system clearly corresponds to a total division of flow. It is known that this type of gradual development of gases is balanced by a loss of load owing to the abrupt change of direction of flow proportional to the square of the initial velocities produced. The Borda turn is another one for loss. When seeking to reduce these losses by making the transition sections too wide, experience shows that the efficiency of the system decreases.

It is ascertained that it is possible to keep them within acceptable limits by partially short-circuiting the latter.

b. Attenuation by Partial Division of Waves

As in total division, we only experimented with partial division up to $l/\lambda = 0.75$. In reality, calculations show that the principle of attenuation by partial division loses its relative advantage (with respect to other modes of HF attenuation) beyond $1/4 \leqslant l/\lambda \leqslant 1$.

When, of the different and successive t paths described above, the shortest or the longest is found to be duplicated by a duct with a clearly greater section, i.e., with a clearly lower impedance than each one of the aforementioned t ducts and providing for passage to a fraction m of the amplitude of pressure P_{ie} (Figure 14), only its compliment $(1 - m)P_{ie}$ is subjected to total division. /44

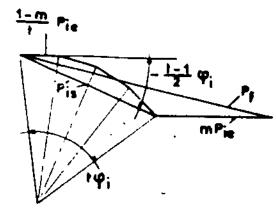


Figure 14.

The flow itself will, furthermore, only undergo a partial division. In this case, the fraction $(1 - m)$ P_{ie} attenuated by total division rejoins the fraction mP_{ie}, phase shifted with respect to the latter by $2\pi l f_i/c$ with absolute value, and is equal to:

$$P'_{is} = \frac{\sin 2\pi l f_i/c}{2\pi l f_i/c} (1 - m)P_{ie}.$$

When the resultant of P'_{is} and mP_{ie} is P_f, it follows that:

$$\frac{P_f}{P'_{is} + mP_{ie}} = \sqrt{1 - 2(k - k^2)\left(1 - \cos \frac{2\pi l f_i}{c}\right)}$$

with:

$$k = \frac{2\pi l f_i/c}{2\pi l f_i/c + \frac{1 - m}{m} \sin 2\pi l f_i/c}.$$

It follows finally for attenuation $\Delta D_i = 10 \log(P_{ie}/P_t)^2$, the expression:

$$\Delta D_i = 10 \log \frac{k^2}{m^2\left[1 - 2(k - k^2)\left(1 - \cos \frac{2\pi l f_i}{c}\right)\right]} \text{ dB}.$$

Plate 21 provides the calculated values taken by this attenuation as a function of l/λ for 3 different m parameters. If m is zero, the attenuation is found by total division. It is only 0 in the case of $l/\lambda = 0$, and is only kept at advantageous numbers when the fraction m, subtracted by divisions from the amplitudes, is not very large. After several fluctuations up to $l/\lambda = 2$, it is practically stabilized for wavelengths less than $l/2$, at values which become larger as m grows smaller.

It is clear that these attenuations can only be produced when the actual gradual development of fractions of amplitudes of HF pressure waves corresponds to that which has been accepted in the preceding. The HF waves involved preexist in the noise spectra at operational modes considered before cut-off of the flow and become transformed into stationary waves when the phenomenon of amplification has the capability of becoming a reality.

The problem of designing the interference moderator consists in searching for the means of carrying out a total or partial division of the flow such that:

a. It corresponds better to the diagrams investigated above;

b. The fraction m is always small;

c. That the initiating section providing passage for this fraction m is of the same order as that of the pipes of the reflection part of the muffler (losses of load).

These conditions imply an input section in the moderator of the fraction to be divided 1 -- m on the order of: /45

$$S = \frac{1 - m}{m} s,$$

s designating the initiating section furnishing passage for the fraction m. Taking into consideration the great practical difficulties to be surmounted on this subject, the idea of superimposing to the above mentioned acoustic mechanism an aerodynamic valve effect turned out to be too expensive.

c. Aerodynamic Valves

Plates 22 and 23 represent vortex valves which we have tested. They are axially fed and give rise to a vortex corresponding to a flow in rV_u = constant, the peripheral component of velocity V_u being minimal directly above the tangential apertures diagrammed in Plate 22 and maximum in the exhaust pipe.

There results from this a flow which is difficult to reverse and the role of valve of this device appears clearly when it is specified that the HF wave train, modulated in LF corresponding to the cut-out noise, becomes visible from the point of view of flow, by a periodic restoration of outside air into the exhaust pipe. When Φ_V (Figure 12), corresponding to s, is zero, m is zero, the division of the flow is total and the vortex integral. The system in question has surprising efficiency so much so that it can by itself be valid as a low-pass. When the loss of load resulting from the muffler system is believed too great, there may take place an "unloading" of the vortex by partial short-circuiting of the latter (m is not equal to zero, partial vortex). This is the case of the moderators of Plates 22 and 23.

It is likewise possible, in order to avoid any rotational flow, to produce diffusion valves (Plate 23). Its principle consists in favoring the flow in the direction of evacuation of gases through the exhaust pipe and to inhibit the necessary expansion in the reverse direction for the return of air. In other terms, the formation of the duct determined in this way is unfavorable for the return of air since it corresponds to a diffusion, i.e., to a compression, and not to an expansion in the atmosphere-muffler direction.

The difference between the two kinds of valves described rests in the fact that with an equal loss of load (on the order of 20 mm Hg in the case of a flow-weak of cold air equivalent to that of the exhaust considered, by approximately 0.060 kg/s) the vortex moderator uses the valve effect to a greater extent than the diffusion moderator. On the other hand, the vortex involves $m \neq 0$, whereas diffusion requires no unloading ($m = 0$). /46

Special Case of Reflection and Interference Low-Pass of the "Free Passage" Type

When s is on the order of the throughputting sections produced, furthermore, in the muffler and when the latter are all aligned, the resultant system is of the free passage type. This is the case of the low-pass of Plate 22 where there is $s = \sigma'' \cong \sigma'$. In spite of the size of this section providing passage to the fraction m, the latter only represents the third of the total intake section of the moderator.

Such a low-pass is characterized by a very low loss of load, but requires a primary muffler with a more widespread HF attenuation and giving rise to a large total (LF and IF) attenuation, especially in the case of a relatively long exhaust outlet pipe.

Single Terminal Reflection and Interference Muffler Without Separate Moderator

The muffler of Plate 24, with 4 reflection stages, with three Helmholtz resonators and a column output resonator forms the final point of our study of single mufflers without absorbents in the rear (Figure 15a).

It corresponds to the electrical diagram below (Figure 15b) which assumes, nevertheless, that the fourth resonator is like the first three ones, a Helmholtz resonator. It is possible for it to be so in reality but under only certain conditions which will be specified below.

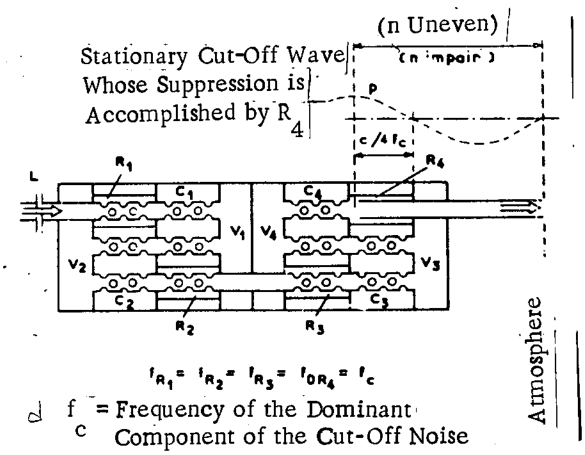


Figure 15a.

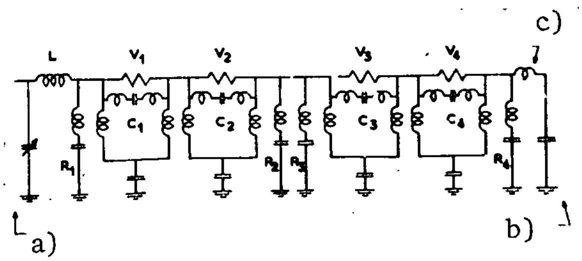


Figure 15b. a, Engine; b, atmosphere; c, exhaust pipe.

It is significant for the difficulty of the problem that all characteristic means of attenuation are employed. Listed in the order in which the exhaust gases pass, these are:

1. A Helmholtz resonator with multiple perforations R_1, at the intake, whose calculated frequency is 1,330 Hz (HF dominance of the cut-off noise);

2. A double-reflection bend C_1 and C_2 with, at the end of each one of these two bends, a widening V_1 and V_2, this system corresponding to the diagram of the muffler of Plate 17 (two symmetrical reflection bends, Figure 11);

3. A resonator R_2 with frequency 1,330 Hz;

4. Cf. 1: R_3;

5. Cf. 2: C_3, C_4 with V_3 and V_4;

6. An output column resonator R_4 with fundamental frequency 1,330 Hz. R_4 is not to be seen on Plate 24.

Comments

1. It has been seen that in the reflection low-passes, the total section of perforations of each one of the parallel pipes (reflection bends) should be of the same order as that of the pipe itself. /48

These perforations should provide passage for a fraction (1 - m) of the flow. The fraction m borrows from the corresponding communication chamber. It is possible to reconcile this diagram with that of Figure 14. The interference effect will become more accentuated when the volumes, background noise for the phenomenon of interference by partial division of flow — become smaller. In the case of the muffler of Plate 24, the interference effect is sought after and the length of the two parallel pipes of each one of these bends C_1, C_2, C_3 and C_4 is such that $l/\lambda = 1/3$ (cf. Plate 21). Furthermore, it is only under this condition that the component with 1,330 Hz is completely suppressed. It reappears, all things being equal furthermore, when $l/\lambda \leqslant 1/4$.

2. A resonator with 1,330 Hz at the input is absolutely necessary. Its casing should be independent from that of the muffler in order to prevent any direct transmission of energy, before attenuation, through the muffler casing.

3. The resonator R_4 is of the column type, on one hand, in order to extend its characteristic attenuation to the upper uneven harmonics and, on the other hand, in order to avoid any reed effect of the HF outlet perforations. When there is a reed effect, the resonator plays its role well at the cut-off frequency but is transformed into a whistle emitting a note absolutely independent of the frequencies of the spectrum components, hence independent of the operating mode, and whose intensity increases with flow.

4. It can be wondered if, in a system such as the muffler of Plate 24, the absorption effect can still be disregarded. Although it is not nonexistent, it is nevertheless very small. It is possible to accentuate it by the fineness of the perforations. With a total equal perforation section, a pipe including a large number of openings of no matter what type, will be more absorbent than with a very small number of holes or merely a simple slot. Furthermore, it can be very quickly confirmed that the damping of the volume resonators always benefits from an energetic subdivision of the aforementioned total section. Nevertheless, in order to avoid their obstruction owing to caking, experience shows that a diameter of 2.5 mm represents a minimum for circular holes.

Characteristics of the Single Terminal Muffler Without Separate Moderator

The muffler of Plate 21 is an excellent low-pass. In spite of its eight stages, of which four are capable of closing losses of load, the sum of the latter is less than that of terminal mufflers with separate moderators. This fact is owing to the size of the fraction of flow taken up by the perforations. In the case of a same outside diameter of muffler (131.2 mm), the type shown in Plate 24 shows 18.5 mm Hg of loss of load when cold with a maximum predicted flow-weight (0.060 Kg/s). On the other hand, the suppression of the worrisome cut-off noise is perfect and offers, in a manner of speaking, a greater margin of safety.

CHAPTER V
SELECTION OF SOLUTIONS

It can be said, in conclusion of our discussion above, that what is concerned here in the whole problem of soundproofing the exhaust is the search for the best solution and principle with respect to the hypothesis of installation of different elements on the automobile, i.e., to define the best compromise of theoretical capabilities. /49

The connection of the high-pass and the terminal low-pass is advantageous if no incompatibility of installation can be interposed (the column high-pass, even mixed, is essentially quite long).

When, in spite of the small diameters of the column high-pass, there can be no question of using this principle and there is an occasion for becoming oriented towards acoustical possible solutions of reflection mufflers. The concept of such systems is, nevertheless, hardly simpler when there is an obligation of installing the main muffler far to the rear of the exhaust pipe. Insofar as concerns the main muffler, it will be remembered that its suitability for installation far to the front of the above mentioned pipe can offer advantages.

Finally, when all ideas concerning a primary muffler have been rejected, it is possible to solve the acoustic problem by designing a single terminal muffler which cannot, however, be more than a mixed low-pass.

As in the case of the intake, it will be seen in Part 4 that the acoustic problem is not the only one to be found within the scope of the building of a modern exhaust.

PART 3
SOUNDPROOFING OF THE INTAKE

CHAPTER I
SPECIAL PROBLEMS WITH THE INTAKE

Part 3 is, like the preceding one, absolutely necessary for understanding /53 of the last part devoted to the filling in of piston engines. Just like for the exhaust, an explanation will follow concerning the essential conditions to be fulfilled in order to solve the problem of soundproofing which is stated and what systems are recommended. It is only after justification of the concept of intake mufflers from the viewpoint of soundproofing that the conditions relating to filling in are described. Automobile drivers have been aware for quite some time of the existence of intake noise. Its intensity is a function of a great number of factors among which we may mention the number of cylinders for a given cylinder capacity. Just as in the case of the exhaust, the 4-cylinder engine is one of the most difficult engines to soundproof insofar as the intake is concerned. Contrary to what occurs in the case of an engine which has four strokes in addition to four cylinders, it does not benefit, as far as noise is concerned, from any overlapping of intake time. Consequently, there is nothing astonishing from the fact that, depending on the conditions of installation on the vehicle and the acoustic characteristics of the latter system, its intake noise can form an important noise source.

Plan of Study

In Part 1, intake noise will be discussed in detail. The problem of soundproofing arises because the influence of its components on the total level is far from slight. The spectra of intake noise show that the low-pitched sounds form the predominant components.

In the absence of any muffler or any air filter the intake duct can be assimilated to one pipe. The latter can be the source of sound waves and resonance phenomena in the same way as an exhaust pipe. In most cases, it is advantageous to group the filter and the soundproofing devices in the same apparatus. The idea of calling upon, in the case of soundproofing of the intake duct, the principle of characteristic attenuation by widening (reflection, cf. Part 1) is completely justified, since the volume in series can chiefly contain the filtering element. It is itself supplied by an intake throat which connects it to the atmosphere.

The acoustic characteristics resulting from such an arrangement, however, cannot give complete satisfaction in the sphere of soundproofing. The possible entering into resonance, taking into account the orders of magnitude, are to /54 be removed as much to prevent them from appearing in the form of noise as to avoid a related phenomenon (of disturbance of carburation).

The required conditions for an intake muffler will be established. It will be a type devoid of defects mentioned above but not complete, nevertheless, removing the intake noise.

The total suppression of intake noise can only occur using more complex mufflers whose design should take into account many factors such as the position of the air intake of the muffler with respect to the other noise sources of the vehicle, the resonance fundamental of the care of the vehicle body, etc. The case of the oil bath filter-muffler for the diesel is special and will not be discussed in order to simplify this report.

Description of the Intake Noise

1. Influence of the Intake Noise on the Total Level

Plate 25 provides an idea of the effect of the load on the total level inside a Peugeot 403 on the roller stand in the laboratory. It is an engine equipped with an intake muffler corresponding to Figure 16a. In order to simplify its designation on the plates, we shall designate this muffler by "intake muffler of Figure 16".

In the above installation, the muffler is connected to the carburetor by a pipe carefully produced from rubber and which has customarily been designated by "intake durite".

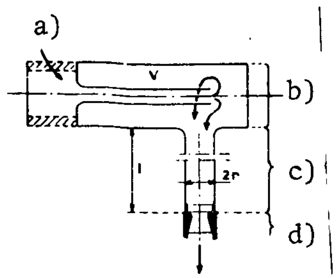


Figure 16a. a, Atmosphere; b, muffler; c, connection (durite); d, carburetor.

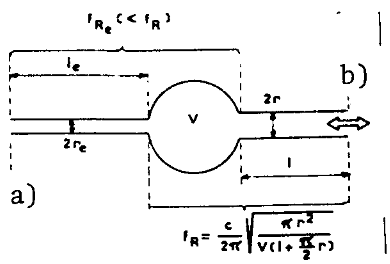


Figure 16b. a, Atmosphere; b, excitation (carburetor).

The recordings of Plate 25 show, for 2 characteristic positions of the microphone, that the total level in the body varies quite appreciably with the load since the "exhaust" noise source is suppressed.

Other recordings made under the same conditions but with the intake noise being plotted at a great distance from the measurement points (Plate 26), show that in the absence of this noise source the other noise sources are either unchanged or suppressed. The level undergoes no variation as a function of the opening of the carburetor valve, i.e., as a function of flow (or load).

It will be seen below that the measuring conditions in a closed room, on a roller stand including above all a trench capable of amplifying or degrading measured levels, do not correspond exactly to those found on the highway. They allow, nevertheless, the making of useful comparative measurements.

2. Spectra and Levels of Intake Noise

We have shown in Plate 27 a spectrum of an intake noise with N $\simeq$ 3,000 rpm. Just as in the case of the exhaust, the high-pitched sounds are common but the low-pitched sounds are predominant and present by far the most problems. The fundamental is $n_c N/30q$ Hz, n_c being the number of cylinders and q the number of strokes of the engine rotating at N rpm. In this way, in the case of the 4-cylinder 4-stroke motor $f_1 = N/30$. The example of the spectrum shown in Plate 27 shows that many harmonics of f_1 are shown, so that the possible excitation frequencies are: /55

$$f_1 = \frac{N}{30},\ f_2 = \frac{N}{45},\ \ldots,\ f_n = \frac{nN}{30}\ .$$

This list should be enough for understanding this chapter but will require a revision for study of filling in. It will be noted, just like in the case of the exhaust, that $f_1 = h_2$, second harmonic of the number of revolutions per second of the engine.

As far as the measured level in the immediate proximity of the source (air intake) is concerned, it may be seen with a muffler such as shown in Figure 16, installed on a Peugeot 403 engine for example, that it passes through a very distinct maximum at a N_R engine operating mode.

Let us consider Figure 16b which depicts the acoustic diagram of the installation of Figure 16a. The different elements of this system (Helmholtz resonators and throats considered separately) can enter into resonance when there is coincidence of their characteristic frequencies with those of the components of the excitation spectra. /56

Taking into account the orders of magnitude involved, this system possesses two main resonance frequencies: those of the Helmholtz resonators R (V, l, r) and R_e (V, l_e, r_e) which make it up:

$$f_R = \frac{c(T)}{2\pi}\sqrt{\frac{r^2}{V\left(l + \frac{\pi}{2} r\right)}}$$

and:

$$f_{R_e} = \frac{c(T)}{2\pi}\sqrt{\frac{r_e^2}{V\left(l_e + \frac{\pi}{2} r_e\right)}}.$$

The source of excitation pressure of R (V, l, r) is localized in the carburetor nozzle which is the source of oscillations of speed and consequently pressure.

In reality, the installation in question is connected to the intake caps of the engine, L being the mean developed length of the duct connecting V to the valves. We have never been inclined to consider the connection of V and L as a Helmholtz resonator.

L behaves as a pipe and its role will be examined subsequently.

It may be ascertained that of the two possible resonances, at frequencies f_R and f_{R_c}, it is the first one which is at the origin of the noise peak at N_R and it can be easily confirmed that

$$f_R(V, l, r) = \frac{N_R}{30} .$$

If l is caused to vary, the noise maximum mentioned above is shifted but can be still seen at f_R.

The resonance of R_e at f_{R_e} cannot occur in the same way for 2 reasons:

1. The resonator R_e is only excited indirectly since it is not directly connected to a source of excitation as is the resonator R;

2. There is produced, in principle, $f_{R_e} \neq f_R$.

Effect of the Intake Muffler on Carburation

The entry into resonance mentioned above, by far the most important because it is very likely caused by the fact of the orders of magnitude involved, it is therefore at the origin of the noise peak mentioned above. But this is not all. It has another effect, much more serious, i.e., a perturbation which is very serious in its consequences for the carburation. The entry into resonance of such a system (Figure 16) is accomplished by a very marked accentuation of fluctuations of speed and pressure directly above the throat of the carburetor nozzle. There results from this a considerable degradation of the operating conditions for which the carburetor has been set up and adjusted.

The size of amplitude of the periodic term which becomes superimposed to the mean discharging velocity under these conditions is such that the air finds it impossible to draw sufficient fuel from the main jet, since the metering of the latter becomes inhibited by an increase in the inertial effect of the gasoline. The direct consequence of this state affairs is to make the carburated mixture lean. The latter situation becomes even more serious when the adjustment of the carburetor at nonidle corresponds to a lower richness level. In other /57 words, the lean mixture which is risked with a system which can enter into resonance is that much more prohibited when the initial adjustment of the carburetor is more economical.

Plate 28 provides the results of measurements of consumption on the engine stand, with comparison of two intake mufflers, one corresponding to the diagram of Figure 17 devoid of unexpected inputs in resonance, the other corresponding to the diagram of Figure 16.

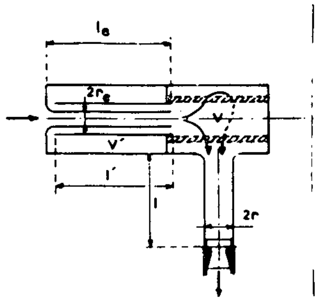


Figure 17a.

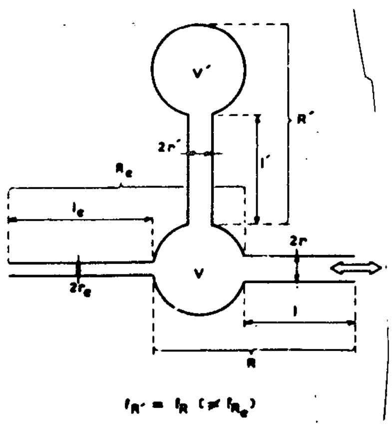


Figure 17b.

The mixture can be so lean that the richness of the carburated mixture just fails to reach the lower limit of inflammability: Plate 29. In these situations, the corresponding power loss is catastrophic.

According to the above, it is possible to state the following fact which has the value of a theorem. Like other conclusions of this type which will follow, we shall classify them:

A. In the case of an intake muffler of an engine with external carburetion by a carburetor, formed by a volume resonator in series, the maximums of sound level and disturbance of the carburetion are the effects from a common cause of which are acoustical in nature — the entering into resonance of the system — and only take place if this latter situation is possible.

Soundproofing Conditions and Good Carburation

1. Carburation

We are now going to show how the two above mentioned defects can be avoided. To do this, it will be enough just to make difficult the entry into resonance described above using a resonator R' having the same characteristics as the resonator R which normally would enter into resonance and whose throat l'

is connected in bypass on volume V (Figure 17a). Figure 17b shows its acoustical diagram. For reasons of design, throat l' and the intake duct l_e of the muffler are concentric and l' has an annular section. l' should be on the same order as l, and the section of throat l' should be equal to that of throat l. There should, in addition, be an identity of characteristic frequencies:

$$f_{R'}(V', l', r') = f_R(V, l, r).$$

which amounts to the same thing as writing the 2 following conditions when T and T' are the temperatures of calculation of the 2 resonators:

(1) $$V\left(l + \frac{\pi}{2} r\right) = V'\left(l' + \frac{\pi}{2} r'\right) ,$$

(2) $$r^2T = r'^2T'.$$

being possible to simplify this last condition to /59

(2') $$r = r'$$

when T = T', usual case of the muffler when R and R' are grouped together in a single body.

These conditions are still valid when l_e is 0. There is obtained, in this case, a muffler without input nozzles such as shown in Figure 18a (acoustical diagram, Figure 18b).

N	Free Intake		Muffler of Figure 16		Muffler of Figure 18	
	w	time	w	time	w	time
(rpm)	(hp)		(hp)		(hp)	
1,000	13	3'07"	12.3	2'55	11.55	2'58"
1,250	16.1	2'27"	15.2	2'09"	15.2	2'21"
1,500	20.5	2'02"	19.3	1'49"	18.25	1'58"
1,750	25	1'45"	25.2	1'51"	23.7	1'40"
2,000	29.2	1'33"	27.8	1'54"	28.1	1'26"
2,250	33.5	1'22"	31.3	1'43"	33.6	1'15"
2,500	37.8	1'12"	36.1	1'30"	37.6	1'10"
2,750	41.9	1'07"	40.5	1'19"	42.3	1'08"

N	Free Intake		Muffler of Figure 16		Muffler of Figure 18	
	w	time	w	time	w	time
(rpm)	(hp)		(hp)		(hp)	
3,000	45.4	1'04"	44.5	1'11"	45.1	1'05"
3,250	47.5	1'01"	48.4	1'04"	47.8	1'04"
3,500	50.7	58"	51.4	59"	51.5	1'02"
3,750	52	54"	53.7	55"	54.4	57 2/5
4,000	56.4	51"	56.1	52"	56.7	53 4/5
4,250	59.4	49"	58.1	50"	58.5	51"
4,500	61.5	47"	59.6	47 1/5	60	49"

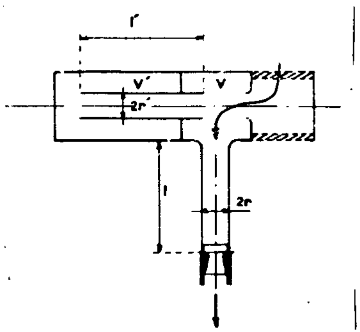


Figure 18a.

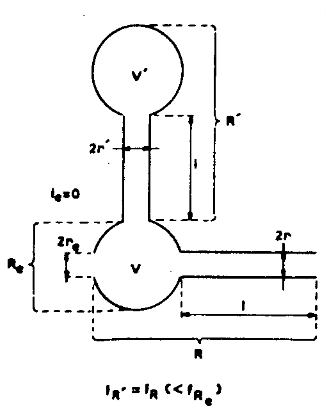


Figure 18b.

When conditions (1) and (2') are complied with, it may be easily confirmed that no starvation phenomenon has been found. The table above shows that with such a muffler without intake throat, the consumption times under full load decrease uniformly as a function of the operating mode, whereas with a muffler corresponding to the diagram of Figure 16, in which both above mentioned conditions are not satisfied, a characteristic impoverishment takes place over a rather wide range of speed and encompasses the critical mode of excitation of resonator R. This impoverishment is reflected by an abnormal variation of consumption times of a given quantity of gasoline (250 cm^3, for example). The zone of disturbance is squared off in the above table (times too long).

The muffler of Figure 18, including no intake throat, is only designed to be protected from the entering into resonance studied in the above. Without intake throat, the volume V is found located in the immediate vicinity of the atmosphere so that the conditions of its optimum position on the pipe representing the intake duct cannot be taken into consideration. Such a muffler, in which only conditions (1) and (2') are fulfilled, remains rather noisy due to this. /61

B. It is possible to design an intake muffler devoid of acoustic properties capable of disturbing carburation without its general efficiency as soundproofing device being satisfactory.

Comments

The extreme case illustrating this fact is provided by the free intake in which case the volume V is infinite (resonance frequency of the Helmholtz resonator corresponding to zero).

2. Soundproofing

In order to satisfy simultaneously conditions of soundproofing and good carburetion, it is necessary to preserve an intake throat with length l_e and radius r_e. The corresponding diagram is that of the standard muffler of Figure 17.

For reasons of space which are as much a handicap for the intake as for the exhaust, the conditions of nonsimultaneousness of resonances of the lengths of pipe preceding and following volume V can be complied with sufficiently in most cases.

In addition to resonances of pipes and the resonance of the volume already studied, it has been stated that there is, in the case of $l_e \neq 0$, a mode of resonance characteristic of the resonator which is called the intake mode R_e (V, l_e, r_e). It is evident that when

$$f_{R_e}(V, l_e, r_e) = f_R(V, l, r),$$

the resonator $R'(V', l', r')$ will have to be designed so as to ensure suppression of the simultaneous resonance from the two resonators R_e and R.

But the dimensional importance needed for the resonator R' is in this case such that difficulties of a practical kind (bulkiness) are inevitably encountered. This is the reason why there is an advantage in stating:

$$f_{R_e}(V, l_e, r_e) \neq f_R(V, l, r)$$

as the third condition for establishing the standard muffler of Figure 17. It may be written explicitly, volume V being common to R_e and to R:

(3) $$\frac{r_e^2}{l_e + \frac{\pi}{2} r_e} \neq \frac{r^2}{l + \frac{\pi}{2} r} .$$

The presence of the intake throat l_e allows volume V to play to the fullest extent its characteristic role of attenuation by reflection (volume in series with pipe, Part 1).

It is likewise necessary for the total attenuation (resonances of pipes) of the muffler, and the condition of nonsimultaneousness of resonance of the parts of pipe preceding (l_e) and following (L) the volume V is quite easy to comply with in general. It is a matter of having: /62

(4) $$\frac{c(T_L)}{4L}, \frac{3c(T_L)}{4L}, \frac{5c(T_L)}{4L}, \ldots \neq \frac{c(T_{l_e})}{2l_e}, \frac{c(T_{l_e})}{l_e}, \frac{3c(T_{l_e})}{2l_e}, \ldots .$$

In this expression, L is the mean developed length of the intake duct counted from the intake manifold to the volume V of the muffler.

The standard muffler of Plate 30 (Figure 17) satisfies conditions (1), (2'), (3) and (4) above and does not show any perceptible noise "peak" although it does not totally suppress the intake noise. Plate 33 shows it installed on the engine.

Comments

1. The law of addition of decibels shows that the total suppression of a noise source is only necessary in some cases (Parts 1 and 2).

2. When l is zero or can be disregarded and $f_R(V, l, r)$ from this fact is very high, the amplification of oscillations of the duct at the level of

the carburated nozzle can become negligible and the resonator R' can be set aside for another role.

When this case (undesirable) occurs, the first two conditions above can be suppressed, the third one being found ipso facto confirmed, since l_e is not zero and there is now occasion for adding:

$$V\left(l_e + \frac{\pi}{2} r_e\right) = V'\left(l' + \frac{\pi}{2} r'\right)$$

and:

$$r' = r_e.$$

3. It is only possible to do without resonator R' when we have:

$$f_R, f_{R_e} \ll \frac{N_m}{30}$$

and:

$$f_R \neq f_{R_e},$$

N_m being the minimum operating mode planned for use of the engine. However, it is practically impossible to fulfill these three conditions.

It is therefore possible to state:

C. In order for an intake muffler to give satisfaction from the viewpoint of soundproofing and carburation of a carburetor engine, it is necessary for it to include a proper correcting device with the same acoustic characteristics as the part of the muffler which would enter into resonance in its absence and an intake throat.

Admissible Loss of Load

Referring to Figure 17 or Plate 30, it can be seen that the loss with abrupt widening, unavoidable with the air intake in the V filtering chamber, is the largest of all. Since the section of volume V is large with respect to the intake section, it is above all the intake velocity U_e which should be limited. It is desirable to keep to $U_e \leq 40$ m/s (maximum flow velocity) in order to limit the Borda loss to $\Psi_m = 10$ g/cm^2, hence the condition: /63

(5) $$\Psi_e = \overline{\omega} \frac{(U_e - U_V)2}{2g} \leq \Psi_m$$

where U_V is the mean velocity in V.

As far as the loss at the outlet of V is concerned, it is absolutely necessary to produce a good contraction coefficient. (1), (2) and (3) above will be used in order to comply with the conditions. It is difficult to have r > 30 mm for a cylinder capacity on the order of 1,500 cm^3.

The dynamic effect of the filter elements has been tested quite completely. However, it is impossible to go into detail on this subject within the scope of this study. The initial velocity U_f of the air passing through the filter elements should be kept less than a certain threshold U_m which is a function of a function of the nature and design of the filters. We shall therefore only provide the maximum admissible figure for this initial velocity in case where the distribution of flow can be made optimum (filter elements as gates): with present day filter materials U_m = 1.5 m/s. Beyond this value, a loss in power begins to be seen (at full load and of the maximum operating mode in question). The condition therefore is stated:

(6) $$U_f \leq U_m.$$

CHAPTER II

TOTAL SUPPRESSION OF INTAKE NOISE

Although the exhaust is by definition the means for removal of gases and its outlet pipe can always be pointed in such a way as to minimize the noise in the ears of passengers, the source of intake noise on the other hand is found more often than not in the engine compartment whose acoustic insulation in the body is far from being easy and is practically never perfect within the low frequency bands. /64

The body of a vehicle represents a rather complex system from the point of view of vibrations. The air in the body can form a resonant system which can be excited in certain operating modes as a function chiefly of the geometric parameters of the body itself. It is quite rare, taking into account the orders of magnitude produced in practice, when the intake noise inside the body fails to find some easy way of transmission to the ears of the passengers at any critical operating mode.

In the case of the Peugeot 403 sedan, the critical frequency is 80 Hz. It has been seen above that, just as in the case of the exhaust, the low-pitched (LF) sounds predominate. The problem of soundproofing the intake is chiefly a problem of the efficient attenuation of the low frequencies. The difficult here is esentially a function of the energy of other sources of noise, i.e., of the level at which they can be corrected. Thus, in the case of a vehicle like the 403, the muffler of Plate 30 forms the best compromise from this point of view. Nevertheless, it is possible to go still further, and it is in this spirit that we have probed deeper into the study of total suppression of intake noise.

Selection of the High-Pass

It can easily be seen, after what has been said on the subject of high-pass devices, that although the Helmholtz resonator forms the ideal device for inhibiting the entering into resonance of a system of the same nature and with the same acoustic characteristics as itself, the uniqueness of its cut-off frequency makes it into a high-pass with characteristic attenuation too localized in the scale of frequencies (in spite of the possibilities for spreadout through damping). Just as for the exhaust, the quarter-wave column high-pass (fundamental) has proven to be superior to all other systems for attenuating the low frequency sounds of the intake noise.

Intake Muffler with Separate High-Pass

For reasons of size, the principle of the quarter-wave closed pipe folded back several times on itself is also to be considered for the intake high-pass. Nevertheless, taking into account the difficulty of the problem of total suppression of intake noise, we have preferred to first of all exploit all the possibilities of the different combinations of column high-pass and low-pass, /65
leaving to one side all considerations of constructive simplification.

It is for this reason that several separate low-pass and high-pass prototypes (not forming part of a single and same muffler body) have allowed us to study the problem stated with more flexibility.

Plate 32 provides an idea of the orders of magnitude arrived at for a fundamental cut-off frequency of 80 Hz. After its passage through the column high-pass folded back on itself once and made of aluminum, the air rejoins a mixed low-pass. The adjective mixed is justified by the fact that the conditions of soundproofing and good carburation studied above have been complied with.

Referring to Plate 29, the excellence of the characteristics of this system can be seen insofar as concerns carburation and power.

Let us now observe the recordings of Plate 33. They relate to a muffler of the type of Figure 16. It is possible to measure the magnitude of the noise going to the intake when conditions (1), (2) and (3) established in the above have not been satisfied. On the other hand, with the system with quarter-wave high-pass (Plate 32), there may be obtained, under the same conditions, the curves of Plate 34. The effect of the intake noise becomes insignificant. Let us examine them more closely: it is possible to wonder from where the apparent anomaly can come, in the rear seats, between 2,800 and 3,000 rpm, speed range for which the residual intake noise has the effect of reducing the total level. It is easy to confirm (by going away from the source of this residual noise), that this is due to an interference effect of the residual intake noise with the "engine noise" component transmitted by the mounts of the latter.

Comments

This phenomenon, which for a given common fundamental frequency of the two noise sources mentioned above assumes a mutual specific phase shift of the latter in order to be perceptible, becomes more obvious as the residual noise in question approaches more closely, in energy, through the one which is transmitted to the ears of the passengers by the engine mounts.

The phase shift (negative) acquired by the residual intake noise with respect to the one which is transmitted to the occupants of the vehicle by the engine mounts is owing to two terms:

1. Phase shift owing to the intake system in its true sense;

2. Phase shift owing to the difference in operation imposed on the waves of the residual noise with respect to the path that those have to travel which originate from the engine through the intermediary of the engine mounts.

It will be noted that the first term contains the effect of negative phase shift of the high-pass itself. The aperture coefficient of the LF column plays from this viewpoint an important role.

For reasons of actual respective impedances of the pipe and the, the usual practice is to select m slightly greater than 1/2 (cf. Part 1) and experience shows that it is only on this condition that it is possible to obtain results as conclusive as those which are featured by the recordings of Plate 34.

In summary, it can be seen that such a system simultaneously solves the problem of soundproofing as well as that of carburetion. We are therefore able, after many confirmations, to state: /66

D. An intake muffler acting efficiently in all components of the noise spectrum, especially in the low-pitched sound area, provides from this fact complete satisfaction from the viewpoint of carburation with the only condition that its loss of load remains low.

Intake Muffler with Integrated High-Pass

Before making a definite conclusion as to the superiority of the column high-pass, we made a last series of tests using Helmholtz resonators. After having succeeded in totally suppressing the intake noise with composite systems of the type described above, it was advantageous to study the possibility of assembling all the necessary acoustic elements into one solid muffler body.

The best results obtained with a high-pass muffler with an integral Helmholtz resonator are illustrated by the recordings of Plate 35. They are far from being equivalent to those which can be attained with a separate quarter--wave column high-pass.

The following level curves (Plate 36) are related to a high-pass solution with an integrated quarter-wave column, folded back twice on itself, but including an input throat with rectangular cross-section and not totally surrounded by that of the resonator at right angles with its connection with V. It can be ascertained that the results are not entirely satisfactory, chiefly below 2,000 rpm and above 2,800 rpm. These defects arise owing to the overlarge transverse dimensions of the (initial) section of the aforementioned intake throat.

The type shown in Plate 37, finally depicts the best solution of the monoblock muffler designed for a total suppression of intake noise. The results obtained with respect to the preceding ones, with this principle, confirm the fact that it is with the high-pass with quarter-wave air column that the best characteristics can be obtained: Plate 38.

On the highway (Plates 39 and 40), the effect of intake noise is not quite as great as on the roller stand but it may be ascertained that the transmission conditions of the engine noise to the body through the intermediary of the engine mounts vary as a function of the load in the direction of an increase of level with the transmitted motor torque. This is the reason why the recording curves under way relative to the muffler of Plate 37 (shown in a fine solid line) are located systematically above the low load level curve. The difference in the ordinance of these 2 curves provide the measurement of the increase in total level owing to the passage from low to full load.

CHAPTER III
SELECTION OF SOLUTIONS

No matter how acute the problem of sounrproofing may be, the intake which /67
occurs on a preliminary basis or after the fact in the study of an automobile, some essential conditions will always have to be complied with. In the most general case, the muffler of the Helmholtz resonator type and with an intake throat corresponding to the conditions of soundproofing and good carburetion established above is the most rational.

In case of necessity it is possible to totally suppress the intake noise and the best high-pass is of the quarter-wave column type. We should like to add, without having shown it within the scope of this study, that the filter muffler with oil bath calls upon the same acoustic principles as the dry filter mufflers. To the conditions stated above on this subject, we should like to add those which should be taken into consideration as far as they concern the concept of the oil bath in its true sense (filtration, suppression of rise of low viscosity oils, operating in an inclined position, etc.).

PART 4

FILLING IN OF PISTON ENGINES

CHAPTER I

FILLING IN COEFFICIENT

Introduction

It was recognized very early in the history of piston engines that the filling in coefficient was one of the main power factors. /71

All things being equal, furthermore, it can be shown that it becomes a factor linearly in the mean pressure of an engine. The flow-weight involved is a function of this. The magnitude of the height employed in a piston engine with respect to that of the corresponding gas turbine only partially favors it since the expansion of the alternating turbine is truncate.

Its gradual development has been possible owing to the continuous efforts in the direction of the increase in compression rate, improvement of combustion, increase in the filling coefficient. These efforts were successful largely because they were helped by metallurgists and petroleum engineers.

The present chapter deals with a completely new aspect of the filling in problem which we have helped exhibit clearly: the acoustic supercharging which has allowed reaching a new stage in the improvement of filling in.

Definition

The filling in coefficient of a piston engine is by definition the ratio of the flow of air actually admitted to that which is created in the time unit by the pistons and measured under the same conditions of reference pressure and temperature p_0 and T_0.

It may be stated for a 4-stroke engine:

$$\rho = \frac{120\ I_{\theta}}{\bar{\omega}_0\ NC_M} = \frac{120\ Q_0}{NC_M}$$

I_0 being the flow-weight admitted, $\overline{\omega}_0$ the specific weight of the air under the above contitions, N the operating mode of the engine with cylinder capacity C_M.

In all the calculations which follow, we shall chiefly deal with the 4-cylinder engine and shall use exclusively 4-stroke engines.

<u>The Advantage of the Filling In Study</u>

The advantage of the filling coefficient concept is considerable for the very reason that it is one of the factors involved in the expression of Serruys providing the mean pressure: /72

$$p_m = \frac{ET_0 p_1 n_3 P_{ci} + \lambda}{10 p_0 T_1 n_2 - V_{m0}} r\eta R_f R_c R_m$$

where:

T_0, p_0 = reference temperature and pressure.
V_{m0} = volume of 1 kg of burned mixture at p_0 and T_0.
p_1, T_1 = temperature and pressure at end of intake.
n_2, n_3 = number of molecules before and after combustion.
P_{ci}, λ = lower calorific value and latent heat of vaporization of the fuel.
r = grade of the intake mixture.
η = theoretical thermodynamic efficiency of the cycle.
R_f = efficiency of form of the actual cycle.
R_c = efficiency of live combustion.
R_m = mechanical efficiency of the engine.
E = equivalency coefficient of work and heat.

The ratio p_1/T_1 corresponds to the specific weight at end of intake:

$$\overline{\omega}_1 = \frac{p_1}{RT_1}$$

directly proportional to the filling in coefficient defined above. Indeed, in the case of outside reference conditions p_0 and T_0, ρ, proportional to I_0, is also directly proportional in this way by virtue of conservation of mass, with $I_1 = \overline{\omega}_1 Q_1$, Q_1 being prescribed by the engine.

It is clear that the goal of all investigations in the field of filling in of engines with atmospheric intake is to provide ρ with the maximum possible value at all operational modes.

The essential causes of imperfection of filling in are three in number:

1. Losses of load in the whole intake system and losses owing to distribution;

2. Heating up during penetration;

3. Nondominant acoustic phenomena.

The second factor will not be taken into consideration within the scope of this study. The first and third, on the other hand, will occupy our attention. A certain irreversible pressure drop is unavoidable upon penetration of the fluid in the cylinder, even in the absence of all loss in the intake duct in its true sense, this penetration being accompanied by the phenomenon of flow with abrupt widening corresponding to the Borda loss. /73

Owing to reasons of design and carburetion, the initial velocity should be kept greater than a specific value along the intake duct, especially in the case of carburetor engines in order to limit the effects of condensation which would unavoidably result from any excessive slowing down of the mixture at full load and at low operating mode.

Although finding the best compromise always requires detailed study of the theoretical or experimental, it is reassuring to observe that the periodic loss of pressure chiefly prevailing directly above the intake valves, hence in the cylinders, can probably be used for filling in the engine owing to the acoustic phenomena which result from it and from which those which are harmful will be removed, at the same time encouraging those which are to be used possibly.

Furthermore, it is possible to compensate, over a specific extent of the velocity range used by the motor, for the effects of inertia (harmful for filling) of the gaseous mass contained in the intake duct. This is successful owing to the device of intake valve closure delay, well-known to automobile experts. This compensation can even pass beyond the scope of reduction to zero of the filling in defect mentioned above in order to be clearly beneficial to resonance. By nondominant acoustic phenomenon, we understand, as much for the intake as for the exhaust, those among them which have an unfavorable influence on filling in, either directly (or resonance: exhaust), or indirectly (outside of the resonance: intake).

Plan of Study

The torque C (N) of an engine is quite appreciably proportional to its filling in coefficient. This is why the filling in curve ρ (N) is one of its most significant characteristics. A filling in curve of an engine with an essentially variable, like that of an automobile, generally has a rather twisted shape and is most often characterized by an apparently random modulation, but well specified as a function of the operating mode.

The attempt is almost invariably made to obtain the best filling in at all operating modes at full load, but it has been conceded for quite some time that it is necessary, in the case of a given engine design and a definition of distribution, to become adapted to the disadvantages inherent in well-known systems.

These disadvantages or filling in defects are what automobile experts commonly call "filling in holes", which characterize any filling in curve ρ (N) whose derivative d ρ/dN changes its sign more than once when at full load the engine passes from minimum to maximum speed.

The goal of this study has been to investigate the reasons for possible disturbances of the filling in curves of piston engines and means of suppressing them.

It will be seen first of all — concerning the problem of filling in measurements that there are grounds for putting the measuring apparatus itself in an area where it will be protected from any entering into resonance (from pipes and capacity).

Two phenomena overlap during intake time which are owing to flow in its true sense with losses of load on one hand, and with an inertia effect from /74 the gaseous column contained in the intake duct on the other hand. The latter gives rise to a modulation of the former: outside of the resonance operating modes (of pipe) the actual pressure p_r at the intake caps appear as the product of pressure p_e resulting from the bringing into velocity with losses beginning from atmospheric pressure p_a by a modulation factor with two harmonics (1 and 2 of the number of revolutions per second of the engine N/60). The harmonic analysis of p_e shows that this pressure contains at least five harmonics (1, 2, 3, 4 and 6 of N/60).

The existence of these five harmonics of p_r allows, with that of the different uneven harmonics of the intake duct (closed pipe on the cylinder side and open on the capacity of the muffler side or atmospheric side in the absence of muffler), the complete qualitative explanation of the shape of filling in curves.

It can be shown that any filling in curve is established beginning from a base curve disregarding all resonance phenomena and determined by the design of the engine (duct and distribution). This curve is amplified and modulated by oscillation phenomena of the pipe and duct caused by the above mentioned harmonics. These quarter-wave oscillations and resonance states which are connected with it (pressure antinodes directly above the intake valve) are at the origin of what can be called acoustic supercharging.

Uncontrolled, these phenomena cause undesirable fluctuations of the filling in that have always been the target of remedial measures. Until this time, the problems which arose had been added to those of disturbance of carburation and intake noise (owing chiefly to the resonances of volume arising from order of magnitude involved) and the exhaust noise discussed in the preceding chapters.

Knowing the causes of these fluctuations, acoustic in origin, allows us not only to remove those turning out to be undesirable, but especially to cause those which are capable of being exploited for the purpose of quantitative improvement [coordinates of ρ (N) and a qualitative (shape) improvement of the filling in curves. An optimization of the filling in is thus possible for a given distribution.

Comments

1. A strict quantitative description of a filling in curve would not be possible unless all data necessary were available which pertain chiefly to the damping of the above mentioned pipe oscillations. These will be examined below. Nevertheless, the exact knowledge of the laws of damping in question and the numerical values of the coefficients involved no longer has more than a minimum interest when the matter is limited to the determination of the different parameters directly affecting filling in. The knowledge of these latter parameters allows the invariable production of the shape of the filling in curve corresponding to the characteristics aimed for, without upper limitation of ordinance.

2. Furthermore, it will be understood why there is no incompatibility between soundproofing of the intake and acoustic supercharging. In reality, as far as the former is concerned, it is sought to suppress the oscillations and volume resonances whereas for the latter, the oscillations and resonances of the pipe are used. These oscillations and resonances are even encouraged /75
by the filtering chamber (volume V) of the muffler, necessary for soundproofing. Let us add to this that the stationary waves which extend from the intake valve to the free V section can cause no increase in the outside sound level (intake noise), although all soundproofing conditions are complied with. These conditions are in no way incompatible with the oscillations and resonance states of the duct connecting the intake valves to V.

Measuring Method

We provide here the measuring method which we have developed because, from our point of view, it is more suitable and more sensitive. It is more suitable because it requires an essentially stationary apparatus and more sensitive because it is differential hence subject to direct redoubt, contrary to the measuring methods by integration hence exhaustion. Moreover, all the acoustic precautions can be taken relatively easy.

The air flow admitted is measured by means of a convergent nozzle followed by a recovery defuser, with conditions p_2 and T_2 at the throat (Figure 19).

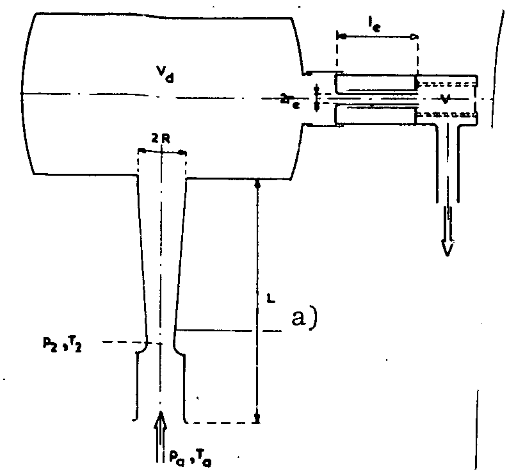


Figure 19. a, Measuring Nozzle.

It is shown that it is very difficult to make a precise correction of the air flow of an engine, measured at p_2 and T_2, to that which would be admitted provided the conditions were p_0 and T_0. The difficulty is avoided by selecting reference /76 conditions such that they are easily reproducible. In the case of a filling in study, a count is taken, during their selection, of the altitude and temperature which can always be maintained during the tests and be reproduced from one test to the next.

This is the same as measuring the filling in coefficient:

$$\rho = \frac{p_2 T_0 120 Q_2}{T_2 p_0 N C_M} .$$

Although the deviation of the manometric liquid is h, the flow-volume Q_2 has the shape:

$$Q_2 = K\sqrt{\frac{h}{\overline{\omega}_2}} .$$

The filling in coefficient is then:

$$\rho = K'\sqrt{\frac{p_2 T_0 \sqrt{h}}{T_2 p_0 N}} .$$

As far as our geographic coordinates are concerned, we have used $p_0 = 1\ \mathrm{kg/cm^2}$, $T_0 = 293°\mathrm{K}$ once and for all.

In reality, it is possible to confuse p_2/T_2 with p_a/T_a, the subscript a designating the atmosphere at the intake of the lead-in duct of the flowmeter. It probably follows that, strictly speaking, the expansion of atmosphere at the nozzle throat is voluntarily small:

$$p_2 = p_a - \bar{\omega}_2 \frac{V_2^2}{2g} .$$

On the other hand, it follows that for the static temperature at the throat:

$$T_2 = T_a - \frac{V_2^2}{2gEc_p} ,$$

c_p being the specific heat of air at constant pressure. Since the ratio p_2/T_2 becomes involved in the expression of ρ, it can be stated:

$$\frac{p_2}{T_2} \cong \frac{p_a}{T_a}$$

and there may be finally written for the filling in coefficient:

$$\rho = K'' \sqrt{\frac{p_a \sqrt{h}}{T_a N}} .$$

The measuring apparatus (Figure 19) is designed such that the intake muffler exhaust the air in a volume representing for it the atmosphere without any degradations of conditions with respect to what they would be provided the muffler was in the free air (infinite outside volume). It is this volume /77 V_d which is fed by the measuring nozzle mentioned above (Plate 41).

Since all the conventional precautions are taken, especially as far as is concerned the efficiency of diffusion, only one single loss remains, for practical purposes, with respect to the atmosphere and that is the one owing to the abrupt widening of the junction of the diffuser with volume and which is unavoidable. The dimension of the system is such that the total loss does not exceed about 10 millimeters of water with maximum flows contemplated.

Precautions of an Acoustic Nature

Since the aforementioned volume plays the role of a buffer, it is relatively easy to produce a flow quite appreciably permanent at the intake, even at the lowest operating modes. Nevertheless, different precautions have to be taken during setting up of the measuring apparatus and it will be seen, as far as the approach duct is concerned, that the AFNOR standards relating to permanent operating modes are no longer valid owing to reasons of an acoustic nature:

1. Any possibility of entering into resonance with the Helmholtz resonator having for volume V_d and for throat l_e (entering throat of the muffler) should be avoided. This amounts to the same thing as writing, when f_M is the lowest excitation fundamental of the engine considered:

$$f_S = \phi(V_d, l_e, r_e, T) < \frac{f_M}{2} . \tag{1}$$

f_S should be less than half of the lowest excitation frequency in order to take into account cyclic irregularities of the engine always possible and observe for practical purposes on all engines (intake tubes and exhaust pipes of varying lengths). In the case of a 4-cylinder, 4-stroke engine, it follows that:

$$f_M = \frac{N_m}{30} .$$

N_m being the minimum operating mode considered, in rpm;

2. The intake frequency of the measuring apparatus, i.e., that of the resonator having as volume V_d and for throat the flowmeter itself, should in no case be able to be seen. When L is the total length of the flowmeter, including approach duct, and R the radius of the connection section of the diffuser with volume, it may be written:

$$f_d = \phi(V_d, L, R, T_a) < \frac{f_M}{2}$$

$$f_d \neq f_S .$$

Of the two possible solutions, there is advantage in using:

$$f_S < f_d < \frac{f_M}{2} , \tag{2}$$

taking into account the fact that the Borda loss at the entering point of volume V_d should be as small as possible;

3. In order to avoid any disturbance of operation of the flowmeter going to possible resonances of pipe L, it should follow that: /78

$$\frac{c(T_a)}{2L} > f'_M \quad (3)$$

f'_M being the highest anticipated intake fundamental. The approach duct being included in L, it is assumed that it cannot be random and in particular equal to the value specified in the AFNOR standards.

We have been led by these conditions to use the following values for 4-cylinder engines which have no more than 21-cylinder capacity: N_m = 1,000 rpm, f_M = 33 Hz, f_S = 10 Hz, f_d = 13 Hz, f'_M = 200 Hz, V_d = 158 l, R = 50 mm, L = 810 mm.

Considerable disturbances can take place when these precautions are not taken. Thus, an overlong approach duct — condition (3) no longer being confirmed — can be the origin of measuring errors amounting to an impressive percentage. We have found errors reaching 200% in cases of entering into resonance of duct L at low operating modes. Such acoustic anomalies can lead to very serious disturbances of operating conditions of the engine which become obvious chiefly through instability in the engine-brake system.

Comments

It should be noted that precautions of this type, intended to avoid all possibility for establishing standing waves in the measuring pipes, should be taken on all occasions when it is a matter of carrying out precise measurements and not only in the case of piston engines.

Soundproofing and measurements of flows which we have had the occasion of studying and carrying out under the most varied circumstances on large tubes (extractors, heat pulses, intake of piston compressors and turbo-superchargers, etc.) would form too broad a subject to be discussed here.

There will be merely noted, in this respect, that the quarter-wave column resonator forms, in the intakes and exhausts of reciprocating engines, the best means for attenuating low-pitched sounds and that, when it is a matter of airing out valid measurements of flow, either for continuous monitoring or with a well-specified study goal, it is absolutely necessary to place the tube or casing being considered in a position where it is protected from entering into resonance.

CHAPTER II
NONAMPLIFIED INTAKE PRESSURE

Two terms become factors in the pressure of nonamplified intake or excitation p_r: /79

1. Pressure p_e resulting from the bringing to speed with losses of load beginning from the atmospheric pressure and which is periodic;

2. A variation of pressure Δp, owing to effects of inertia from the gaseous column contained in the intake duct, likewise periodic.

These two magnitudes which will be established disregarding all resonance phenomena, are at the origin of the above and form the pressure of excitation p_r (α). It is the harmonics of the latter which create the states of resonance which will be clearly exhibited.

Pressure p_e

In the absence of any amplification phenomenon, pressure p_e prevailing directly above the intake manifold is equal at any instant to that which prevails in the corresponding cylinders owing to laws governing flow in a duct with extremely abrupt widening, as is the case of passage from the intake valve of a 4-stroke motor into the cylinder.

It can be calculated as follows, by assuming that only the instantaneous velocity of the piston has an influence on the velocity of air in the intake duct. This hypothesis amounts to the same thing as disregarding the effect of the variable aperture of the valve around the intake time, and is found to be justified by the existence of closure delay at intake and by the orders of magnitude involved.

Let there be a vehicle below (Figure 20), in which l is the length of the crankshaft and r the half-stroke of piston P.

It follows for the piston stroke, when $r/l = \lambda$:

$$x = r(1 - \cos \alpha) + l(1 - \sqrt{1 - \lambda^2 \sin^2 \alpha})$$

and for its velocity, with $\alpha = 2\pi t/T = \omega f$, T being the period, ω the pulse and t the time:

$$v = r\omega \sin \omega t + l\lambda^2\omega \sin \omega t \cos \omega t(1 - \lambda^2 \sin^2 \omega t)^{-1/2}$$

and finally in the case of the pressure corresponding to this velocity, creat- /80
ed by expansion beginning from the atmospheric pressure p_a with losses of load:

$$p_e = p_a - K\sigma^2 \frac{\omega}{2g}\Big[r^2\omega^2 \sin^2 \omega t + 2r\omega^2 l^2\lambda^2 \sin^2 \omega t \cos \omega t(1 - \lambda^2 \sin^2 \omega t)^{-1/2}$$
$$+ l^2\lambda^2\omega^2 \sin^2 \omega t \cos^2 \omega t(1 - \lambda^2 \sin^2 \omega t)^{-1}\Big]$$

in which:

$\sigma = \frac{S_r}{S_c}$ = ratio of discharging sections of the piston and the intake cap or valve head;

K = numerical coefficient taking into account all the elements of loss of load of the complete intake duct.

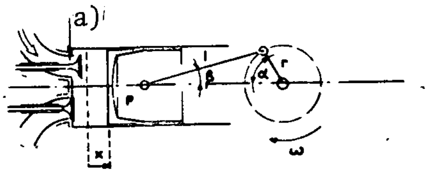


Figure 20. a, Intake Manifold or Valve Head.

It is the strict expression of the pressure p_e in the case of the figure above in the absence of any phenomenon capable of modifying its amplitude. The harmonics will be sought for the purpose of their later utilization.

Development of Pressure p_e as a Fourier Series

Let:

$$C_1 = K\sigma^2 \frac{\omega}{2g}\omega^2 r^2,$$

$$C_2 = 2K\sigma^2 \frac{\omega}{2g}\lambda\omega^2 r^2,$$

$$C_3 = K\sigma^2 \frac{\omega}{2g}\lambda^2\omega^2 r^2.$$

It follows for the pressure that:

$$p_e(\alpha) = p_a - C_1 \sin^2 \alpha - C_2 \sin^2 \alpha \cos \alpha(1 - \lambda^2 \sin^2 \alpha)^{-1/2}$$

$$- C_3 \sin^2 \alpha \cos^2 \alpha \ (1 - \lambda^2 \sin^2 \alpha)^{-1}.$$

Considering the very small value of λ^2 generally found in modern engines, it is possible to validate a simplified hypothesis consisting in considering the factors between parentheses of the second term as being equal to unity, thus giving:

(1) $$p_e(\alpha) = p_a - C_1 \sin^2 \alpha - C_2 \sin^2 \alpha \cos \alpha - C_3 \sin^2 \alpha \cos^2 \alpha.$$

The calculations of the Euler integrals (or, moreover, a more direct calculation) allows putting this expression into the following form: /81

(1') $$p_e(\alpha) = p_m + A_1 \cos \alpha + A_2 \cos 2\alpha + A_3 \cos 3\alpha + A_4 \cos 4\alpha$$

knowing that:

(2) $$p_m = p_a - \frac{C_1}{2} - \frac{C_3}{8}$$

and in which:

$$A_1 = \frac{C_2}{1},\ A_2 = \frac{C_1}{2},\ A_3 = \frac{C_2}{1}, \text{ and } A_4 = \frac{C_3}{8}\ .$$

This development can only have an indicative value, considering that in all strictness, it is only valid when there is no discontinuity. Indeed, the pressure $p_e(\alpha)$ does not effectively cover the entire period but nevertheless extends over a large fraction of the latter taking into account the actual values of AOA and BFA.

There is no doubt that only precise measurements allow the clear exhibit of all harmonics involved. A considerable experimental work has allowed us to observe that, in addition to the four harmonics supplied by the reasoning above, others exist no matter what type of intake duct is considered, especially harmonic 6.

Variation of Pressure Δp

To this pressure $p_e(\alpha)$ overlaps a quantity Δp owing to the inertia of the column of air contained in the intake duct. Just as in the case of $p_e(\alpha)$, a determination will be made of its expression beyond any resonance state. Under these conditions, the mass to be taken into consideration is that of the above-mentioned column with developed length L_{SA}. The oscillation of pressure owing to acceleration (beginning of the piston stroke) and with slowing down (end of stroke) of this mass has the form: /82

$$\Delta p = -\sigma \frac{\bar{\omega}}{g} L_{SA} \frac{dv}{dt},$$

p being the instantaneous velocity of the piston established above. There will then be stated for the density:

$$\frac{\bar{\omega}}{g} = \frac{p_e}{gRT}.$$

If the terms with λ^2 are still disregarded, there is produced for the accelration:

$$\frac{dv}{dt} = r\omega^2(\cos \omega t + \lambda \cos 2\omega t),$$

hence:

(3) $$\Delta p(\alpha) = -\frac{\gamma\sigma L_{SA} r\omega^2}{c^2}(\cos \alpha + \lambda \cos 2\alpha) p_e(\alpha).$$

Actual Pressure of Excitation p_r

This pressure results from:

$$\Delta p(\alpha) = p_r(\alpha) - p_e(\alpha).$$

Placing this with (3):

(4) $$p_r(\alpha) = \left[1 - \frac{\gamma\sigma L_{SA} r\omega^2}{c^2}(\cos \alpha + \lambda \cos 2\alpha)\right] p_e(\alpha).$$

The actual pressure, apart from the resonance, results therefore from the pressure owing to flow $p_e(\alpha)$, periodic and multiplied by a modulation factor with two harmonics. The nature of this factor which is owing to the above mentioned inertia effect, becomes obvious at the beginning of the intake time by an attenuation and at the end of the intake time by an accentuation of pressure $p_r(\alpha)$.

Base Curve

The gaseous column contained in the intake duct represents a damped oscillating system. It can be considered, counted from the intake valve head to the filtering capacity of the muffler (or quite simply to the atmosphere in the absence of the muffler) as a pipe closed at one end (valve) and open at the other. Let L_{SA} be its developed length.

It has been seen that pressure p_e contains at least the harmonics h_1, h_2, h_3, h_4, h_6 of the number of revolutions per second N/60 of the engine. On the other hand, the modulation factor owing to the inertia of the air column may be broken down itself into two harmonics h_1, h_2. The duct L_{SA}, to be considered as a pipe, is the source of oscillations whose resonances with 1/4 λ, 3/4 λ, 5/4 λ..., at frequencies $H_1 = c(T)/4\ L_{SA}$, $H_3 = 3H_1$, $H_5 = 5H_1$, ..., correspond to some characteristic operating modes at harmonics h_i of pressure above.

It is therefore possible to define a curve of minimum ordinates of the coefficient of filling in such as it would be provided the above-mentioned oscillations were not amplified. The formation of this base curve provides a certain advantage since it is this curve which determines to a certain extent the general aspect of the actual filling in curve. It is this curve which is overlapped by the acoustic and inertial effects which will be studied below. Three limiting conditions supply three characteristic points for this curve.

First Characteristic Point

Even apart from the resonances, the phase shift of each of these oscillations cannot be disregarded. When it is negative, it justifies the delay with closure of the intake (RFA). It has been known for quite some time that the RFA encourages the filling in at high operating modes then encourages the equalization of pressures prevailing in the intake manifold, on the one hand and in the corresponding cylinders, on the other. Nevertheless, at low operating regime, the RFA has a consequence of a partial compression of the load admitted.

Let α_{RFA} be this delay of intake valve closure. The corresponding compression stroke of the piston is, with the preceding notations: /83

$$\alpha_{RFA} = r(1 - \cos \alpha_{RFA}) - l(1 - \sqrt{1 - \lambda^2 \sin^2 \alpha_{RFA}}).$$

With velocity infinitely small (N = 0) it follows that, at the end of intake ($\alpha = \pi + \alpha_{RFA}$) a fictitious or calculated pressure:

(5) $$p = p_\alpha \left(\cos^2 \frac{\alpha_{RFA}}{2} + \frac{1}{2\lambda} - \frac{1}{2}\sqrt{\frac{1}{\lambda^2} = \sin^2 \alpha_{RFA}}\right)$$

supplying the ordinate of the intake pressure curve with 0 velocity (point A, Plate 12). It has been seen, concerning the definition of the filling in coefficient ρ, that this latter is proportional to the pressure for a given temperature at end of intake. The following reasonings will relate to the pressures and will not refer to ρ except for the actual filling in curves.

Second Characteristic Point

Although the operating mode $N \cong 0$ is characterized by maximum compression, it is possible to try and clearly exhibit another one, N_0, which would be characterized by the reduction to zero of this compression at the atmosphere. Let us consider by definition that, with this operating mode N_0, there is perfect equalization of the pressures of both sides of the intake valve considered and that the compression flow likewise is reduced to 0 at the valve.

This amounts to the same thing as assuming that with N_0:

1. The maximum pressure is reached in the cylinder in the case of $\alpha =$ $= \pi + \alpha_{RFA}$ at the same time as in the duct L_{SA};

2. The gaseous column L_{SA} is compressed by inertia to zero flow at the valve;

3. The final pressure in L_{SA} is equal to the atmospheric pressure p_a. This last condition supplies the ordinate of the point sought after: this is the same as that of point A [equation (5)].

The 2 first ones allow the clear exhibition of the effect of the RFA on the abscissa N_0 of different waves. Of all the arguments that can be made beginning from the above-mentioned hypotheses and which arrive at appreciably equivalent results, we shall only offer the most simple.

Since the flow is 0 at the valve, there is adiabatic compression in the cylinder from $\alpha = \pi$ to $\alpha \doteq \pi + \alpha_{RFA}$. On the other hand, the compression by inertia of the air column under these conditions causes the involvement of half of the mass of air considered. Finally, the form of the increase in pressure in the duct L_{SA} is to be reconciled with that of equation (3). It is thus possible to state:

$$\left(\frac{2r}{2r - \alpha_{RFA}}\right)^{\gamma} = \frac{p + \Delta' p(L_{SA}/2)}{p} .$$

Since the pressure p is shown in the numerator of the expression of $\Delta'p$, its exact value does not become involved. In addition, since $\Delta'p$ is no longer a function of α beyond the low neutral point ($\alpha = \pi$), since the flow is 0 at the valve beyond which value it is possible to state: /84

$$\left(\frac{2r}{2r \quad \alpha_{RFA}}\right)^{\gamma} - 1 = \frac{1}{2}\frac{\gamma\lambda\sigma L_{SA}\omega^2 r}{c^2}$$ [signs illegible]

hence:

(6) $$N_0 = \frac{30\sqrt{2}}{\pi} c\sqrt{\frac{\left(\frac{2r}{2r \quad \alpha_{RFA}}\right)^{\gamma} \quad 1}{(1 \quad \lambda)\gamma\sigma L_{SA} r}} .$$ [signs illegible]

This is the reduction to zero operating mode of compression which, in the case of (5), defines point B (Plate 12).

Third Characteristic Point

Even apart from all resonance (except for 0 velocity), there is a phase shift (negative) of the actual pressure with respect to the excitation pressure. It becomes greater as the ratio of the vibration imposed on the natural vibration considered becomes larger. The RFA, whose value is selected so as to limit the losses of filling in by expansion in the case of $0 < N < N_0$, therefore tends to become more and more insufficient when the operating mode of a given engine is increased. On the other hand, the amplitude of Δp increases as the square of the velocity. Taking into account the orders of magnitude, it is possible to say that at high operating modes the pressure curve at the end of intake tends to become reconciled with that of the mean pressure p_m. It becomes reconciled even more quickly when the RFA is smaller (Plate 12).

It is therefore possible to specifically say that the base curve, which is to some extent the skeleton of the actual filling in curve, as will be seen subsequently, becomes more spread out as the RFA increases in magnitude. Indeed, N_0 becomes greater, and the equalization of pressures on one side and the other of the intake valves is carried out more easily at a high operating mode when the RFA is greater.

All things being equal, furthermore, an engine with a slight delay of intake closure has a more bulging filling in curve which, nevertheless, decreases more swiftly than that of an engine with a large RFA which is maintained much longer at high values when the operating mode increases. Small RFA values (on the order of 30°) are produced for private motor vehicles, more often than not used at low operating speed, and large RFA values (on the order of 60° and more) in the case of racing vehicles called upon to supply considerable high-speed torques.

The base curve ABC (Plate 42) is therefore made up of two parts: AB = zone of compression, BC = zone of drop. This latter value is a function simultaneously of the RFA and the sum of the losses of load of the system of intake tube and the muffler.

CHAPTER III
ACOUSTIC SUPERCHARGING

Phenomenon of Acoustic Supercharging

To this base curve may be added the amplitudes of possible oscillations /85
mentioned in the beginning of the preceding paragraph. They enter into resonance every time that one or several of the harmonics $h_1, h_2, h_3, h_4, h_6, \ldots$, coincide with one or several of the resonance harmonics of L_{SA}, i.e., $H_1 = C(T)/4L_{SA}$, $H_3 = 3H_1$, $H_5 = 5H_1, \ldots$, corresponding to vibration modes in $1/4\lambda$, $3/4\lambda$, $5/4\lambda$, ... The following considerations will be limited to the three resonance harmonics and to the five above mentioned excitation harmonics.

When consideration is given, the excitation mode resulting from the above, it may be ascertained that qualitatively and quantitatively h_2 is the largest pressure harmonic.

Let N_{SA} be the operating mode at which the resonance frequency H_1 corresponds to the harmonic h_2 of the pressure. It then follows that:

(1) $$H_1 = \frac{c(T_{L_{SA}})}{4L_{SA}},$$

(2) $$h_2 = \frac{N_{SA}}{30},$$

hence:

(2') $$N_{SA} = 7.5\frac{c(T_{L_{SA}})}{L_{SA}}.$$

This last relation provides the nominal acoustic supercharging operating mode. The resonance of L_{SA} has the effect of increasing the final intake pressure, since it is understood that the RFA plays in this respect a favorable role since the phase shift of the oscillations is negative (Figure 21). The optimum value of the RFA (from this viewpoint) is a function of the order of the excitation harmonic considered.

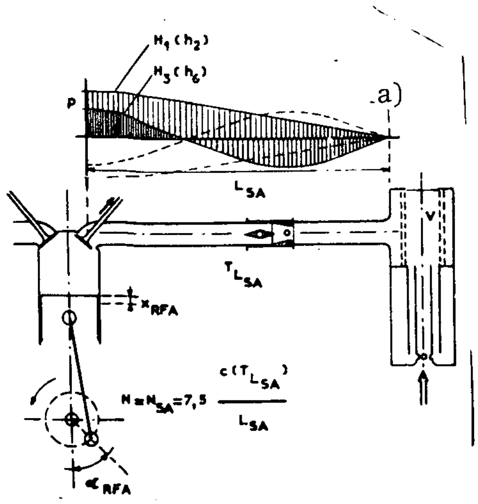


Figure 21. a, Free Section (Volume V of the Muffler or Atmosphere).

The table of Plate 43 lists the cases of coincidence of harmonics of excitation and resonance for L_{SA} for the different characteristic operating modes which can occur. Plate 44 groups the main types of modern intake ducts diagramed for 4-cylinder 4-stroke engines. The corresponding arrangements for a 2-cylinder engine can be easily deduced from the above. The /86 single cylinder engine is still of the type C. This table diagramatically illustrates the differences which can occur between the various types of intake ducts insofar as the distances between valve heads are concerned.

The relative amplitudes at uneven harmonics have a sign different from that used in the beginning ($\alpha = 0$) and at the end ($\alpha = \pi$) of the intake stroke, but are positive in the case of $\alpha = \pi$, both in the expression of $p_e(\alpha)$ [equation (1')] as in the expression for the modulation factor [equation (3)] for h_1.

The result is that, taking into account values of A_1 and A_3:

1. A_1 has the tendency of being reduced to 0 at the intake valve head of a given cylinder with $\alpha = \pi$ by the amplitude of the harmonic of the same order relative to the intake time of the neighboring cylinder, when this time follows the first (the four cylinder case). This tendency becomes more obvious as the distances between valve heads becomes smaller (types A and A' with 4 cylinders in line and A as flat-twin);

2. A_3 considered separately undergoes the same effect from its counterpart which succeeds it in the neighboring valve head;

3. The 2 uneven harmonics h_1 and h_3 of amplitudes of A_1 and A_3 mutually /87 oppose one another; the absolute values of the latter are equal. They reduce each other in the case of $\alpha = \pi$;

4. h_1 nevertheless continues to exist because it is likewise contained in the modulation factor.

This mutual partial reduction to zero does not, on the other hand, take place with even harmonics. It can generally be stated that the increase of pressure at the end of intake ($\alpha = \pi + \alpha RFA$) is less favored by resonances coming from uneven harmonics (h_1 and h_3) then by those which are created by the even harmonics (h_2, h_4, h_6). These latter ones are, with h_1, at the origin of the acoustic supercharging effects, i.e., of the possible increase of pressure at end of intake which is added to the base curve.

Comments

The listing of the properties of the above harmonics is deduced from the numerical values of the A_1 amplitudes and can appear doubtful at first sight. It is, nevertheless, perfectly justified owing to the fact that only the effects of harmonics h_2, h_4, h_6 (and h_1) have been revealed after experimenting with more than 160 different assemblies, with the exclusion of any effect of h_3, however small it might be. This latter is only included for the sake of form in Plate 43 (operating modes 2/3 N_{SA} and 10/3 N_{SA}).

Actual Filling In Curves

In the following, the pressure concept will be abandoned to return to that of filling in. A considerable amount of work on the stand allowed us to carry out an extremely concentrated series of practical observations on the engine, on one hand, and the preceding theoretical deductions, on the other hand. When any random filling in curve is taken into consideration (Plate 45), it can be seen that there results an amplification, at characteristic operating modes (Plate 43), of the base curve owing to acoustic supercharging. It is noted that, unfortunately, uniform amplification is not involved.

Also, any random curve includes peaks and values. The juxtaposition of filling in peaks and valleys can be very bothersome and give rise to prohibitive fluctuations of $\rho(N)$. This can also cause a lack of torque corresponding to overaccentuated peaks as well as inhibiting travel with ignition, the automatic function of the carburetor and metering in the case of injection carburetion. There is therefore an advantage in suppressing these peaks of the $\rho(N)$ curve on one hand, and to profit as much as possible from the capability of amplifying $\rho(N)$ by acoustic supercharging.

Filling In Peaks

The curve of Plate 46 was taken from a 7 rate hp, 4-cylinder engine having an intake duct of type A and supplied with an intake muffler of type (Part 3, Figure 17, Plate 30) satisfying the four conditions of souddproofing and good carburetion. We shall disregard for the moment the disturbances which can arise from the exhaust, the latter being assumed uninvolved.

This curve includes three peaks, at operating modes 1,550, 2,325 and 4.650 rpm, corresponding to the resonance modes of the L_{SA} following: H_1 excited by h_6, H_1 excited by h_4, and H_1, H_3 excited respectively by h_2 and h_6. The filtration chamber V of the muffler plays the role of free section (open end of L_{SA}). Plate 17 illustrates the actual arrangement of such an intake duct. Experimental confirmation can be made that volume V (Plate 46) plays the role mentioned above better when the four conditions relating to the concept of the standard muffler referenced above are better complied with. /88

When the intake section of L (Part 3, Figure 17) is completely free and when L_{SA} (some of L and the mean distance from the intake valve head considered to the carburetor intake, cf. Plate 46) is awarded the desired length in order to locate N_{SA} at an operating mode which is considerably lower than in the assembly described in Plate 46 (N_{SA} = 1,825 rpm, for example), it is confirmed that it is possible to produce an extremely energetic supercharging effect corresponding to resonance modes H_1 and H_3 of L_{SA} respectively excited by h_2 and h_6 (Plate 48).

This logical result is favorable for the accentuation of filling in at low operating modes so sought after in small cylinder capacities. It is therefore possible to state that acoustic supercharging is easier to employ at low operating modes than at high operating modes and becomes more intense with increasing perfection in the acoustic diagram produced to cause it, i.e., when L_{SA} is more free at the proper end.

Filling in Valleys or Holes

These can have either separate or simultaneous origins:

1. They result from the juxtaposition of peaks (Plates 45 and 46);

2. They are the consequence of effects of acoustical counterpressure with exhaust.

Plate 48 illustrates the latter case. The explanations which can explain this phenomena are similar to those which have been reported above on the subject of the intake duct.

Nevertheless, there is occasion for taking temperature into account to a much greater extent. This is the parameter which can be considered as practically constant for the intake.

We have taken into account the effect of acoustic counterpressure with the exhaust within the scope of this study as a consequence of the observed impossibility of normally supercharging both operating modes N_2 and N_1 given by:

(3) $$\frac{N_2}{30} = \frac{c(T_{N_2})}{4L_1},$$

(4) $$N_1 = \frac{N_2}{2}\sqrt{\frac{T_{N_1}}{T_{N_2}}}.$$

in which L_1 designates (Plates 48 and 49) the mean developed length of the exhaust valve heads with the free section materialized by a primary muffler with large total attenuation (cf. Part 3). The experimental confirmation of this phenomenon is carried out by location of the nominal characteristic operating mode N_{SA} of acoustic supercharging at which the effect sought after /89
becomes minimal. In the assembly in question, the supercharging H_1 (h_2) and H_3 (h_6) is minimum at N_2 = 3,150 rpm in the case of L_1 = 71 cm (Plate 49). Taking into account the temperatures, (3) and (4) are confirmed in this way, knowing the length of l, hence of L_{SA}, the least favorable to supercharging by h_2 and h_6: this is certainly a case of counterpressure of acoustic nature

caused by the entering into resonance with $1/4\lambda$ of L_1 (antinode of pressure directly above the exhaust valve head).

Suppression of Disturbances of $\rho(N)$ Owing to the Exhaust

Referring to Part 2, it will be noted that these entries into resonance are:

1. More accentuated when the volume of the primary muffler shown schematically in Plate 48 is larger, i.e., that its total attenuation is greater;

2. Situated at operating modes which become lower as length L_1 becomes greater;

3. Situated at operating modes which become higher with corresponding increase in temperature T_L.

It has been seen in Part 2 that there is an arrangement of the primary muffler in which a good total attenuation can be produced at the same time as a characteristic (HF) advantageous attenuation. This is the one resulting/ from a reduction of L_1 to the minimum: Plate 50. It is possible in this way to succeed in increasing N_2 in such a manner that the new operating mode N_2' clearly passes beyond the boundaries of the diagram ($N_2' > 6{,}000$ rpm). As for the resonance with N_1' owing to harmonic 4 of this operating mode [$N_1'/15$, cf. relation (4)], it is found in this way sufficiently reduced to pass unseen for two obvious reasons:/

a. $N_1'/15$ is too high to be able to give rise to amplitudes comparable to those illustrated by Plate 49;

b. Just as for the intake pressure, the excitation amplitude of harmonic 4 of the exhaust pressure is clearly lower than that for its harmonic 2.

The correction for the Peugeot 404 exhaust therefore forms an especially ingenious solution for the total of the problems studied thus far:

1. Total and characteristic attenuations which are advantageous insofar as the soundproofing of the exhaust is concerned;

2. All the acoustic pressures being reduced to 0 in the forward volume in question and in the exhaust muffler in its true sense, the result is that

the counterpressure is found to be reduced to that which is owing to uniform and nonuniform losses of load connected with the delivery, excluding any effect of acoustic counterpressure;

3. The infrastructures of the bodies tending to become lower and flatter and, owing to this fact, making almost impossible the installation on the pipes of any intermediate mufflers no matter how flat they are, the solution of the volume with bypass in the direct proximity of the valves is the only one which allows profiting from the acoustical advantages of the two-muffler system (Part 2); /90

4. The problem of corrosion (above all oxidation) in the heated state is found to be solved ipso facto; no delivery passing through the aforementioned volume which is nevertheless brought very quickly to a temperature sufficient to stop any condensation on its walls.

Suppression of Filling in Valleys Owing to the Intake Duct

Let us now take up this well-known problem which had been believed unsolvable until quite recently with the arrival of the Peugeot 404. We have found three solutions for it:

1. By Superposition of Two or Several Filling In Curves Having Peaks and Valleys

This solution, developed on the 7 hp, 4-cylinder engine, includes a special muffler satisfying the functional conditions set up in Part 3 and whose filtration volume V (Plate 51) is connected to the carburetor by two rubber pipes (durite) l_1 and l_2 respectively determining two mean developed lengths of complete duct L_{SA} such as there results from the better distribution of characteristic operating modes of acoustic supercharging over the whole range of speeds.

This principle of highly selective superposition of several curves (two being enough, in general) has a greater range of use than is generally assumed at first sight.

2. By Optimum Damping of Resonance Phenomena of L_{SA}

This is the most simple solution and the one which is put into practice in series on the Peugeot 404 engine. It consists in spreading out the resonance phenomena of L_{SA} by only slightly decreasing their intensity (Plate 52). This effect is produced by partially placing in the atmosphere the forward end of throat l' of the resonator R' (V', l', r') of the intake muffler. The volume V is found in this way deprived of a certain part of its quality of free section, although the four nevertheless fulfilled.

Plate 53 shows the position of the special damping perforation provided for this purpose. The rubber pipe (durite) connecting the muffler to the carburetor has a length such that:

$$N_{SA} = 2{,}200 \text{ rpm.}$$

In this way, there is produced a filling in curve with a considerable advantage for a passenger vehicle with a relatively small cylinder capacity (high torque at low operating modes: 13 mkg at N_{SA}, absence of hollows).

3. By Continuous Acoustic Supercharging

It consists theoretically in producing: /91

$$N_{SA} = N, \tag{5}$$

N covering the entire range of speeds which is the same thing as providing L_{SA} with the following law of variation as a function of N (Figure 22):

$$L_{SA} = 7.5\frac{c(T)}{N} . \tag{6}$$

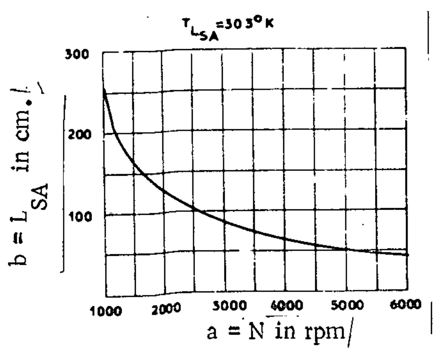


Figure 22.

This method, which is to be reconciled with the first solution (above), will form the subject of a separate report when it is complete.

Optimization of the Curve Shapes for a Stock Automobile Engine and a Racing Car Engine

It has been seen that the base curve determines the general aspect of the filling in curve in that it is itself a function of the RFA and of the curve of mean pressures. It is also true that its amplification owing to acoustic supercharging is essentially a function of the design of the intake duct. In reality, no detail concerning this latter fails to reflect on the value of $\rho(n)$. It will be noted that:

1. In the case of a relatively small RFA, there corresponds a compression and a compression zone which is smaller than with a considerably larger RFA. Consequently, in the first case, the filling in curve intersects more quickly the mean filling in (corresponding to p_n) beyond N_0, operating mode for reduction to zero of the compression, than in the second case;

2. All things being equal, furthermore, the drop of $\rho(N)$ beyond N_0 becomes more accentuated when the sum of the losses of load existing in the system of the one or more intake ducts becomes greater. The illustration of this fact will be found in Plate 54, since both curves compared were recorded under strictly identical conditions on two engines which only differ by their cylinder capacity. In both cases, the distribution and intake duct as well as the RFA are identical. It follows that in this case, cf. relation 2 of Chapter II, Part 4): /92

$$\left(\frac{C_1}{2} + \frac{C_3}{8}\right)_{7ch} < \left(\frac{C_1}{2} + \frac{C_3}{8}\right)_{9ch}$$

and consequently:

$$p_m,7ch > p_m,9ch.$$

Nevertheless, when the actual curve system is considered, it may be noted that there is a limit to the increase of discharging sections between carburetor and intake valve heads (and between distributor and intake valve heads in the case of the injection engine), since the excitation amplitudes A_1 (equation D-II-1') are themselves proportional to coefficients C_1, C_2, C_3.

When it is considered that, in addition to the filling in alone, the distribution requirements (reduction to the minimum of deviations in the mixture grade coming to the cylinders in carburetor engines), it is noted that the plot of the duct connecting the carburetor to the intake valve heads should correspond to the best compromise between the requirement for maintaining in suspension the droplets of gasoline and that for reducing to the minimum the corresponding losses of load. It is necessary, for this, to ensure a minimum discharging velocity at the lowest full load operation.

This second reason for an upper limit of sections is clearly no longer valid in the case of the engine with direct or indirect (into the valve heads) injection.

The characteristics of a stock engine are all the more appreciable as its filling in curve reaches higher values at low operating modes and when its filling in curve shows less disturbances and when the sum of the losses of load of its intake duct in its system is more reduced.

The operating curves for a racing engine should result from the care for producing a maximum specific power, taking into account the formula required. Its torque should decrease slowly as a function of the operating mode, and, in order for it to be kept at high values up until the maximum predicted speed, there will be acceptance of a large expansion for $N < N_0$. The type of duct C (Plate 44) is now universally recognized as being the best one for racing vehicles. The optimal length of these separate pipes is then, according to the above:

$$L_{SA} = 7.5\frac{c(T)}{N_M} , \tag{7}$$

N_M being the maximum predicted power operating mode.

The characteristics required for a privately operated vehicle imply therefore a relatively low RFA (on the order of 30° to 45°), whereas 60° truly represents a minimum value for a motor whose basic quality should be its specific power (ch/I).

Comments

It is worthy of note that the actual value of RFA of an engine is determined, in practice, by a conventional value of the lift of intake valves below which it may be considered that this latter is definitely closed. In reality, the cam profiles are customarily established for the purpose of producing a slight variation in the accelerations of the valves and it has been found suitable for this reason (and others) to define, in the case of a cam an inclined plane upward and an inclined plane downward, also called planes of silence. Taking into account the great differences in temperature to which the kinetic chain of engine distribution is subjected, a total well-specified minimum play j_1 is required, when cold, between the movable part (valve) and the engine part (cam). The RFA taken into consideration in the above is defined by taking into account a conventional play when cold with the valves $j_2 > j_1$. /93

Role of the Intake Muffler

It has been seen above that the capacity V containing the filtering element of the muffler which acoustically plays the role of volume in series on a pipe (characteristic attenuation by widening of the section and total attenuation, cf. 1st, 2nd and 3rd parts) as well as the role of Helmholtz resonators R (V, l, r) and R_e (V, l_e, r_e) (cf. Part 3), is the more or less free section of the L_{SA}, intake duct. This results from the developments above where the role of the free section of V becomes better assured as the frequencies of pressure oscillation possible of the latter become more removed from the harmonics of supercharging.

If consideration is made of the standard muffler of Plate 30 (Part 3, Figure 17), in which conditions 1, 2, 3, 4 and 5 of Chapter I, of Part 3 are confirmed, it may be noted that the oscillations of the V pressure of maximum amplitude possible are located at frequency f_{Re} (V, l_e, r_e), hence the condition of freedom of section V (1):

$$f_{Re}(V, l_e, r_e) \neq \frac{(2K - 1)c(T_{L_{SA}})}{4L_{SA}}$$

where K = 1, 2, 3 ..., according to which it is desired to achieve an effect of maximum supercharging at resonant frequencies of $L_{SA}H_1$, H_3, H_5, ... (cf. Plate 43). Nevertheless, in practice (orders of magnitudes) this condition becomes:

$$(8) \qquad f_{Re}(V, l_e, r_e) \neq \frac{c(T_{L_{SA}})}{4L_{SA}},$$

even if the system of capacity V of duct L_{SA} (of radius r) formed a Helmholtz resonator, which would not be the case for an intake duct of type C (cf. Plate 44), there would always be:

$$f_R(V, L_{SA}, r) \ll \frac{c(T_{L_{SA}})}{4L_{SA}}$$

and there would be arrival at the same condition of freedom of section V.

The damping of R' can be produced either by exposure to the partial atmosphere of volume V' or by exposure to the partial atmosphere of throat l' (Plates 52 and 53), or by the two arrangements simultaneously. /94

Only the second solution is used. Any exposure to the atmosphere of volumes of Helmholtz resonators — who, in order to play their role, are the seat of pressure oscillations — would have as a consequence an emission of noise. The principle of exposure to the partial atmosphere of the throat of resonator R' (V', l', r') is highly flexible and allows an experimental development which is both swift and precise. For practical purposes, there is a limitation to a minimum number n_m of orifices which are necessary and sufficient to cause $d\rho/dN$ to only change sign one time as a function of N. An orifice diameter (5 mm, for example) is used which will only cause a number of orifices n to vary. It should follow that:

$$(9) \qquad n = n_m.$$

To relations (1), (2'), (3), (4), (5) and (6) of the first chapter of Part 3, it is therefore advisable to add the two conditions (2') and (8) of Chapter 3 of Part 4 as well as (9) as conditions for establishing the intake system.

Plates 55 and 56 show a filter-muffler with oil bath constructed according to the principles stated above. Its acoustic diagram is the same as that of the standard muffler of Plate 53 with dry filtration.

Designed in a very short time, no revision has been required. During the experimental phase of the study, it was enough, provided the least disturbance of curve $\rho(N)$ was ascertained, to slightly modify n number of orifices of the damping perforation of R'.

Role of the Exhaust

Insofar as filling in is concerned, the ideal case is that of the two-muffler exhaust. The first muffler, simple volume, can be located at distance L_1 from the valve heads such that:

$$N_2' = 7.5\frac{c(T_{N_2'})}{L_1} > N_M, \tag{10}$$

N_M being the maximum velocity of utilization [cf. relation (3) above]. This is the case of the exhaust of the Peugeot 404 engine (Plate 57). When an exhaust consistent with the diagram of Plate 50 cannot be constructed, the most advantageous principle along the lines of filling in is that of the collector with separate outlets. The separation of outlets has the effect of dividing the amplitude of acoustic counterpressure with resonance by a number which is simultaneously a function of the number of cylinders and the value of the individual sections of the valve heads with respect to the one or more exhaust pipes in their true sense. Plate 58 depicts a racing car exhaust solution with separate outlets grouped two by two. The primary muffler (with interference by partial division of amplitudes) produces a scattering effect which has the goal of reducing the initial velocities in the greatest possible fraction of the total length of the pipes. All the terms of losses of load are found in this way to have been practically divided by four downstream from the primary muffler (reduction of counterpressure owing to flow).

Comments

1. The reader has probably noted that, insofar as concerns the exhaust, /95
we have limited our study to the causes of their remedies. The above concerned

in no way effects of suction, advantageous or not, such as the Kadenacy effect. We carried out a laboratory study of the capabilities of diffusion, during which we found it possible to ascertain that it only becomes advantageous when it can be employed quite close to the engine. Carried out at the level of the far end of the exhaust outlet, it is only rational when the initial velocity can be sufficiently reduced. This is rarely the case for reasons of bulk and cleanliness.

2. What is termed the "intersection of vàlves" is the result of a delay in closure of the exhaust intended to encourage the evacuation of gases as well as a result of a moving forward of the intake aperture intended to increase the total time of aperture of the latter. Shafts with racing cams always have a rather pronounced intersection of valves as well as having, above all, a considerable delay in intake closure.

Effect of Filling in On Specific Consumption

It is well-known that it is difficult to obtain, with a carburetor, curves of specific consumption which are entirely satisfactory at all operating modes.

Although in actual use it may be convenient to have available a certain excess of mixture at very low speeds (from 1,000 to 1,500 rpm approximately), the latter is often over-abundant and, in most cases, its reduction leads to a prohibitive impoverishment between 2,000 and 1,000 rpm. This problem therefore concerns the automatic character of the carburetor. It is not only possible, in this case, to act on all the functional parameters of the carburetor (operating jets and their arrangement), but also on the air itself.

In the case of the above mentioned difficulty, it is possible, by increasing filling in between 1,000 and 1,500 rpm, to reduce the specific consumption to these operating modes without degrading in any manner whatsoever the operation predicted for the carburetor at higher flows.

The table below confirms this method which we have carried out on a 1,618 cc cc engine (Peugeot 404) with Solex 32 PBICa (adjustment 25, 135, 185) which, as is known, includes an automatic emulsion system. For each operating mode N we provide two filling in values ρ to which correspond the specific comsumptions c_s which become increasingly smaller as the filling in improves:

N (rpm)	1,000		1,100		1,200		1,300		1,400		1,500	
ρ (%)	84.5	86.2	85.3	87.2	86.2	87.6	87.2	88.1	88	88.8	88	88.9
c_s (g/ch/h)	290	288	303	289	310	287	300	275	284	260	260	250

The imperfection of automatic operation, seen with many variations on practically all carburetors, is one of the aspects of the carburation problem in general which has provided specific encouragement for the development of fuel injection. In the case of the latter, it can be said that the specific consumption is less than that of the carburetor engine in all points of operation in the work-operating mode area, knowing that a strict comparison of the two carburation systems (carburetor and injection) can only be made with two identical engines insofar as concerns their thermodynamic definition, their cylinder capacity as well as their distribution. /96

Comments

1. The principle of carburation by injection into the valve heads allows acceptance of a compression rate greater than that which can be produced with a carburetor, for a fuel with a given octane rating, and, furthermore, the injection engine is adapted to a poorer air-gasoline mixture than that of the carburetor engine (simplification of distribution and condensation problems). Finally, the convergent-divergent nozzle including the air valve in the injection engine can be designed such that the loss of load which it causes is considerably less than that of the corresponding carburetor.

The result is that the injection engine is characterized by a network of specific iso-consumptions which is more favorable than that of the corresponding carburetor engine.

2. All conditions established in the above — with the exception of the one which shelters the carburation (by carburetor) from disturbances — remain valid for the injection engine.

Summary of Conditions for Designing Intake and Exhaust

In conclusion of all the preceding, a listing can be made of the following conditions for designing intake and exhaust mufflers.

a. Intake

First, the case of the standard intake muffler of Plate 53 will be considered and it will be assumed that the base curve will be determined by the distribution. The nine following conditions should be fulfilled:

1. Two conditions of minimum loss of load: 5 and 6 of Chapter I, Part 3;

2. Four conditions of soundproofing and carburetion: 1 to 4 of Chapter I, Part 3;

3. Three acoustic conditions of filling in: 2', 8 and 9 of Chapter III, Part 4.

b. Exhaust

The case of the exhaust resulting from the association of two reflection mufflers will be considered. The conditions to be complied with are as follows: /97

1. Condition of minimum loss of load (cf. Part 3, Chapter III): initial sections of the pipe as large as possible, design of mufflers;

2. Conditions of soundproofing: complimentary attenuations of both mufflers;

3. Acoustic conditions of filling in: 10 of Chapter III, Part 4.

We hope that we have made a sufficiently clear explanation of the whole problem of filling in, dominated as it is by phenomena of acoustic nature. The latter have always existed owing to the fact of the orders of magnitude involved in high-speed piston engines. But it is only after having exhibited them clearly and analyzed them that it has been possible to consider their control and finding the best solution by carrying out acoustic supercharging. It is now clearly understood how to optimize filling in, i.e., its adaptation to the desired characteristics. This capability may be added to all progress made in the last few years in the state-of-the-art of the piston engine which is far from having finished its brilliant career.

PLATES

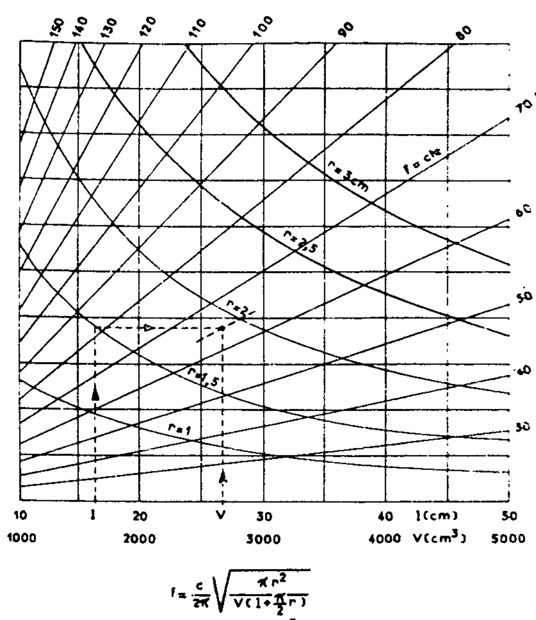


Plate 1. Helmholtz Resonator for Air. T = 288°K.

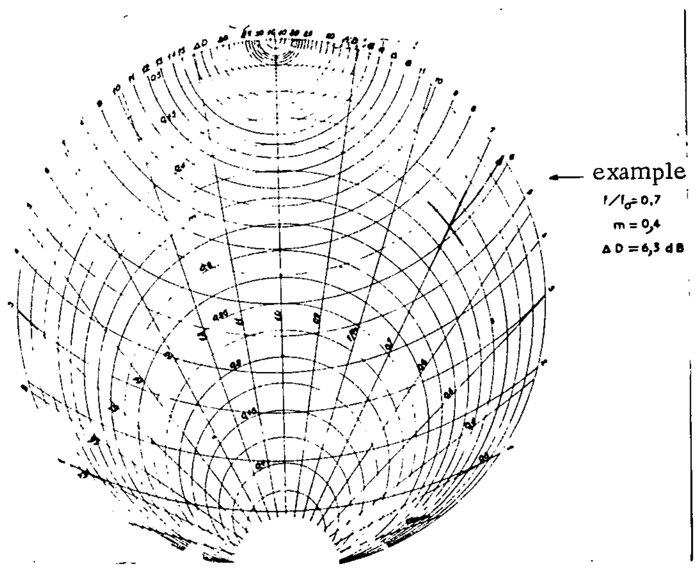


Plate 2. Attenuation by Interference. Nomogram providing

$$\Delta D = 10 \log \frac{1}{1 - 2(m - m^2)(1 - \cos \frac{f}{f_0}\pi)} .$$

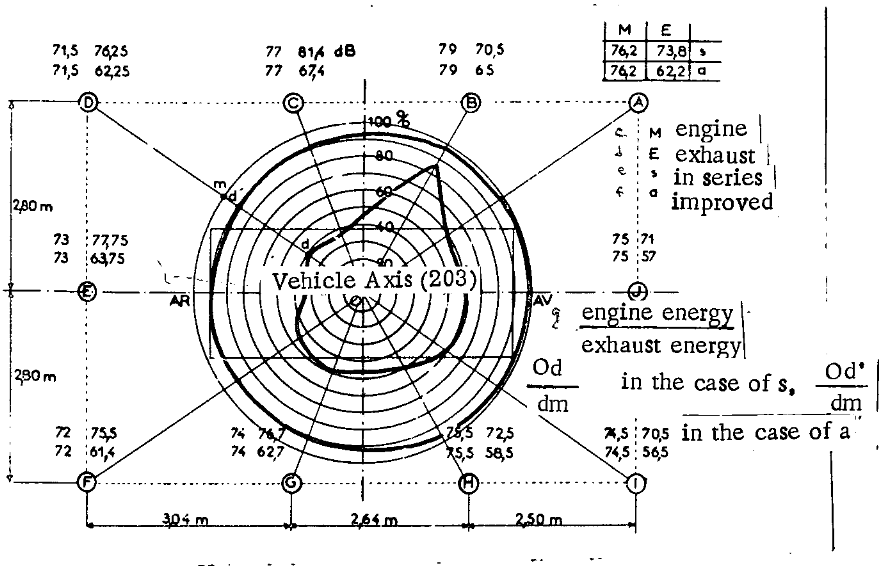


Plate 3. Components of Outside Noise with No Load at Points ABCDEFGHIJ (N = 2,250 rpm).

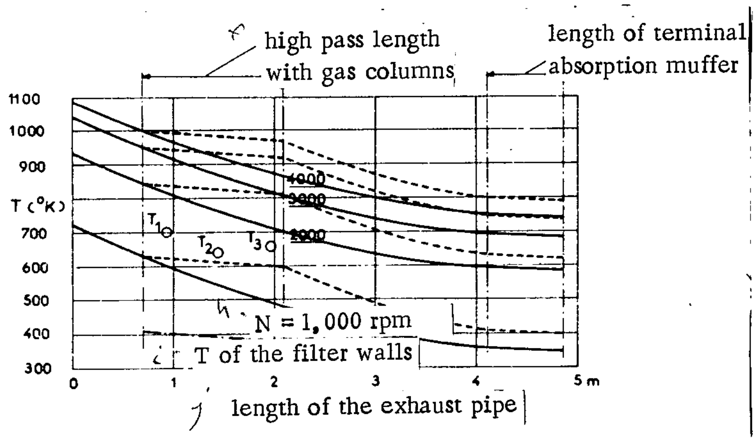


Plate 4. Main Temperatures of Exhaust in Load (203).
——— exhaust 203 in series;
------ exhaust with triple high-pass (with three gas columns);
T_1, T_2, T_3 calculation temperatures of the gas columns.

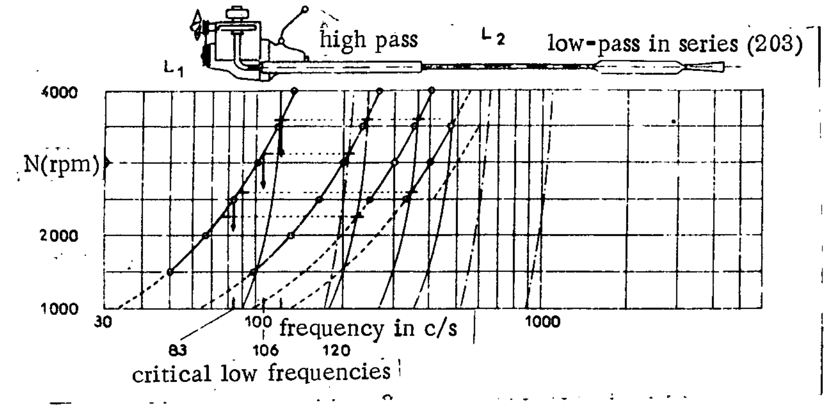


Plate 5. Graphic Determination of the Calculation Frequencies of a High-Pass with Three Gas Columns.
- – —— - engine frequencies;
- – – – – – L_1 resonances;
- ———— L_2 resonances.

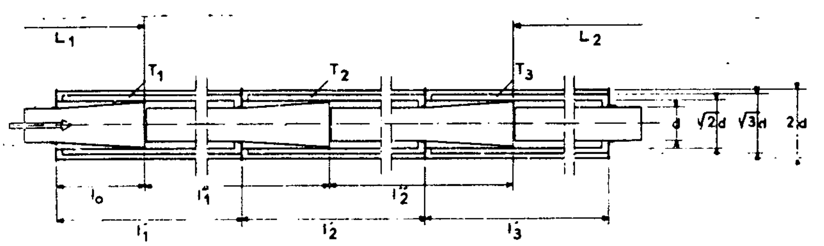
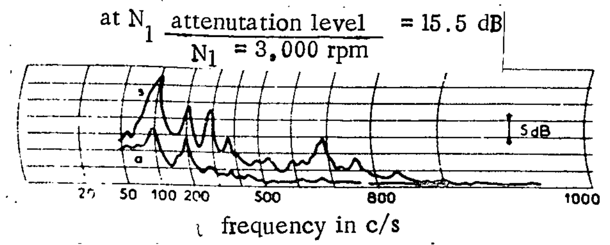


Plate 6. High-Pass with Three Columns of Gas for Exhaust 203 (Very Schematic Cross-Section)

$$3l_2' - l_0 = \frac{c(T_1)}{4f_1}, \quad 3l_2' - l_0 = \frac{c(T_2)}{4f_2}, \quad 3l_3' - l_0 = \frac{c(T_3)}{4f_3}$$

Note: In the case of f_1, f_2, f_3 and T_1, T_2, T_3 cf. Plates 4 and 5. For all explanations, see text.



s - exhaust in series;
a - exhaust with high-pass.

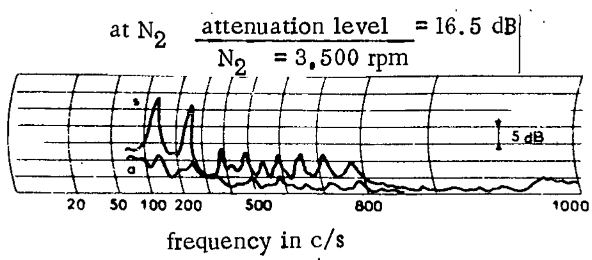


Plate 7. Attenuation of the Exhaust Noise Under No Load Obtained with Exhaust 203 in Series by Attachment of a High-Pass with Three Gas Columns.

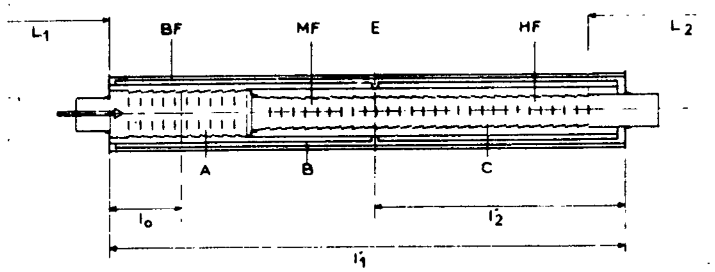


Plate 8. Mixed High-Pass with Division of Flow for Exhaust 203 (Very Schematic Cross-Section).

$$3l_1' - l_0 = \frac{c(T_1')}{4f_1}, \quad 3l_1' - 2l_2' - l_0 = \frac{c(T_2')}{4f_2}$$

A - diffuser;
B - columns of gas;
C - mixer.

For all explanations: cf. text.

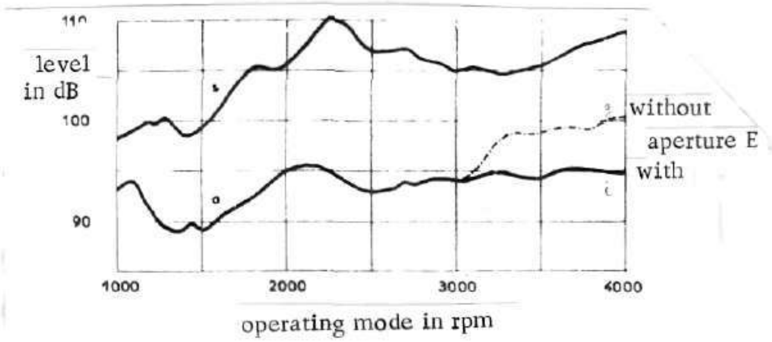


Plate 9. Noise Attenuation of Exhaust Under No Load Obtained with 203 in Series by Attachment of Mixed High-Pass with Division of Flow.

Microphone at 25 cm from the outlet and at 45° from the direction of the gases.

s - series
a - with high-pass

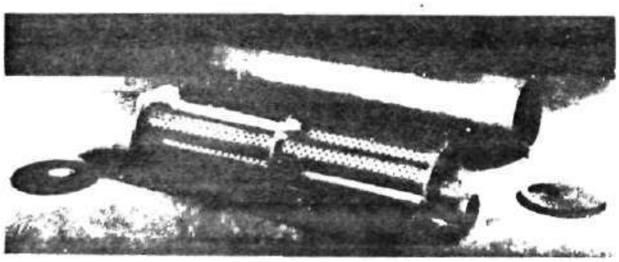


Plate 10. Rear Reflection Exhaust Muffler for Peugeot 403.

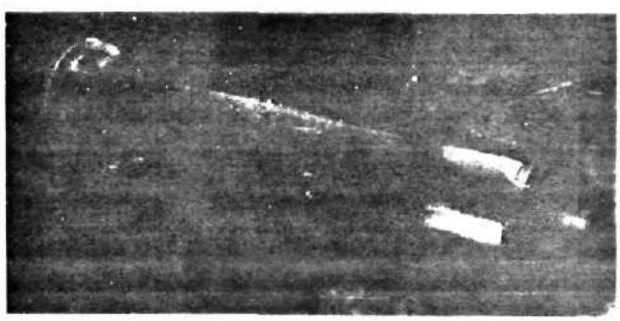


Plate 11. When the Primary Muffler is a Sufficiently Effective Low-Pass, the High-Frequency Perforations of the Rear Muffler Can Be Replaced by Slots (Narrowing of the High-Frequency Output Band). Design prototype.

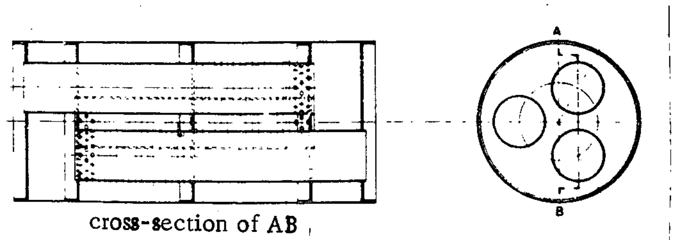


Plate 12. Double Output (HF) Stage for Two-Muffler Reflection Exhaust.

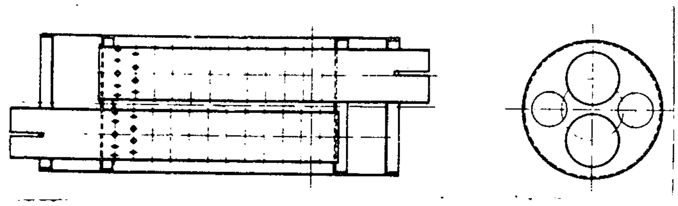


Plate 13. Primary Exhaust Muffler for Two-Muffler Exhaust, Characteristic Action: IF and HF

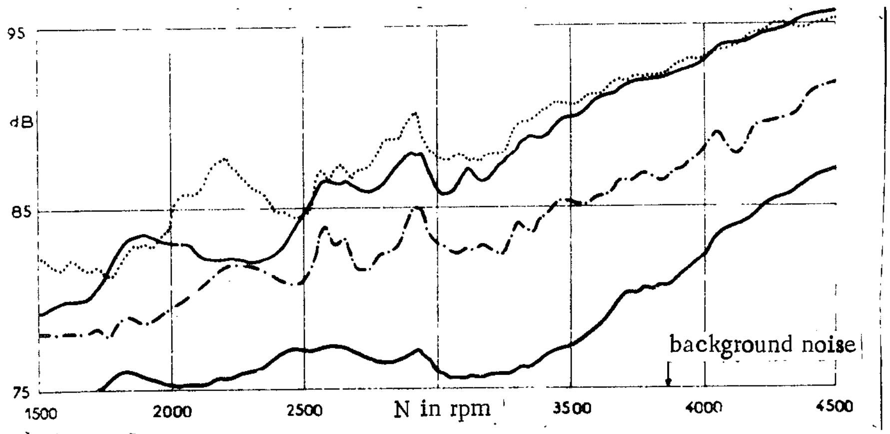


Plate 14. Levels of Exhaust Noise. Recordings carried out on the stand with 403 engine under partial load after suppression of other noise components with 4 different exhaust installations with the same total length, adjustment on the stand for 40 hp at 3,500 rpm. N is made to vary through the intake.

.... at absorption muffler (with Rear);
——— upper AV low-pass and absorption muffler;
——— lower 403 in series (two reflection mufflers, the main one with Rear);
-·-·-·-· single reflection muffler with Rear absorption moderator

Microphone 1 meter from the outlet and 90° from the direction of the gases; decibelmeter: flat.

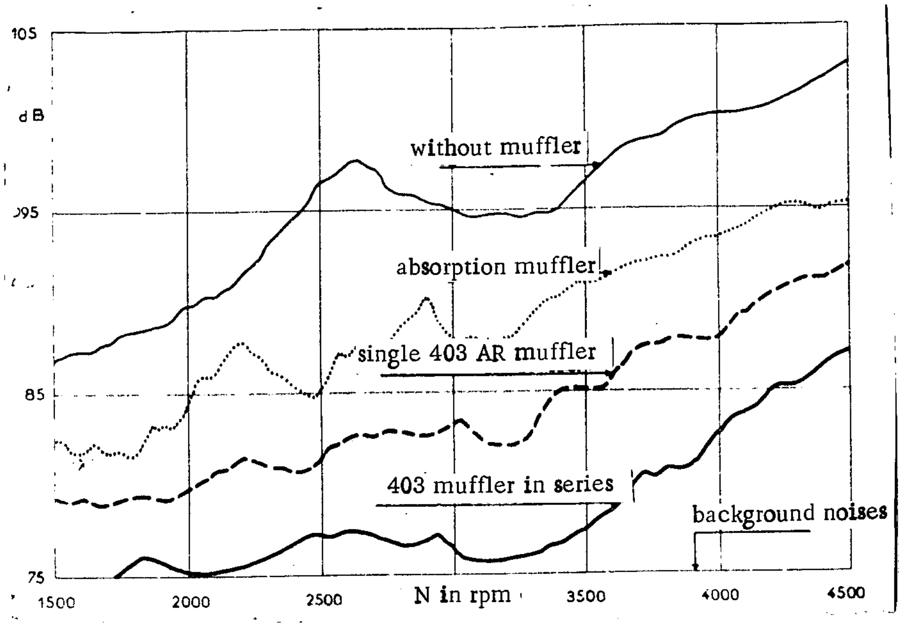


Plate 15. Total Level of Different Exhausts with Same Length (403). Measuring conditions: see preceding plate.

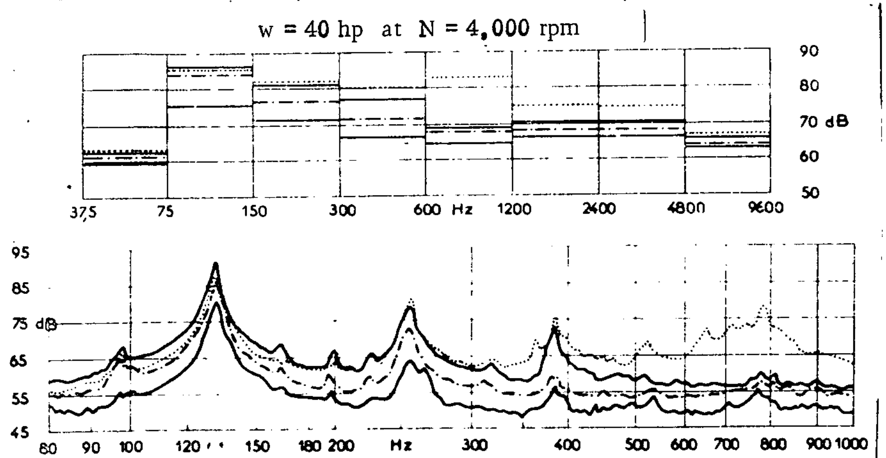


Plate 16. Spectra and Analyses by Bands of Exhaust Noises. Recordings carried out on the stand with the 403 engine under partial load after suppression of other noise components with four different exhaust installations with the same total length.

.... at absorption muffler (with rear);
———— upper Rear low-pass and absorption muffler;
———— lower 403 in series (two reflection mufflers, the main one with rear);
-·-·-·-· single reflection muffler with rear absorption moderator

Microphone at 1 meter from the outlet and 90° from the direction of the gases.

Plate 17. Primary Exhaust Muffler with Slots. Characteristic effect: IF, HF, total effect LF

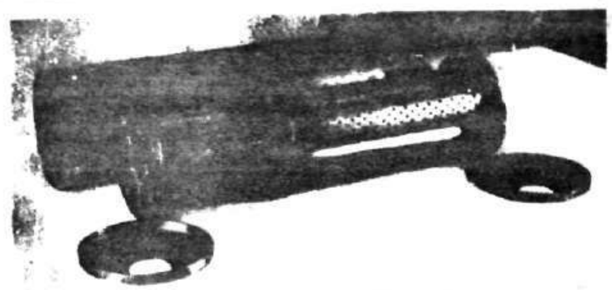


Plate 18. Rear Reflection Four-Stage Exhaust Muffler Two reflection bends and two Helmholtz resonators (HF, on at outlet).

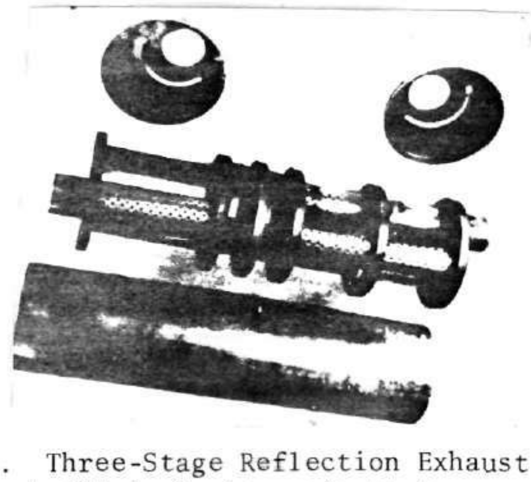


Plate 19. Three-Stage Reflection Exhaust Muffler for Peugeot 404 in Series, cf. text.

Plate 20. Crackling Sounds with Abrupt Cut-Off of Gases Under No Load at 4,000 Rpm in the Absence of Moderator. Polar oscillogram showing the appearance and frequency of the maximum amplitude wave initiated with each closure of an exhaust valve; the spot rotating at the speed of the engine: two excitations per rotation for a 4-cylinder engine.

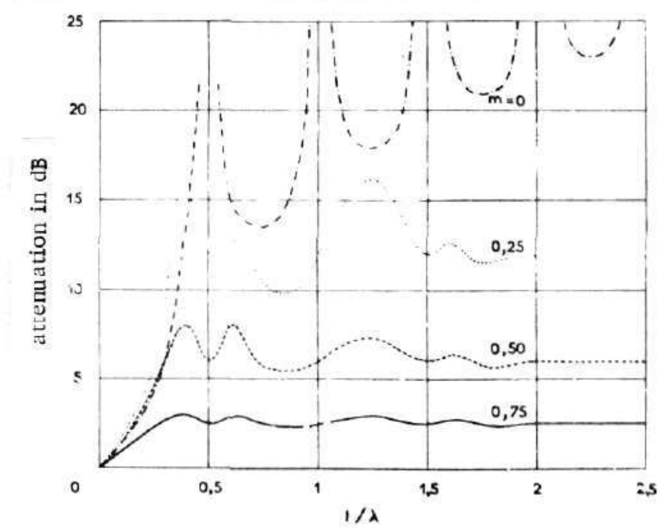


Plate 21. Attenuation by Total or Partial Division of the Flow (Interference).

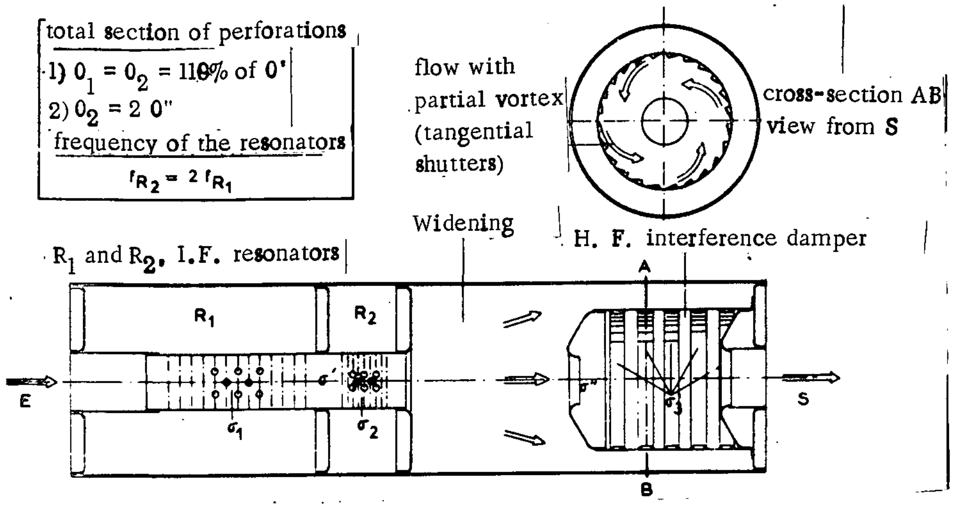


Plate 22. Terminal Reflection and Interference Low-Pass.

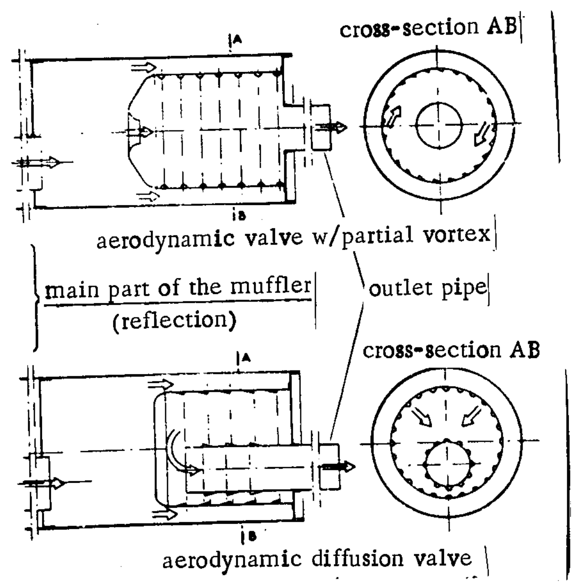


Plate 23. Interference Dampers.

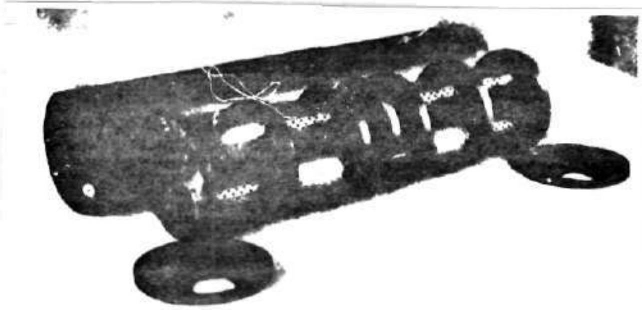


Plate 24. Reflection Exhaust Muffler for Installation at the Rear End of the Pipe Without Muffler or Forward Damper. For makeup and operation: cf. text.

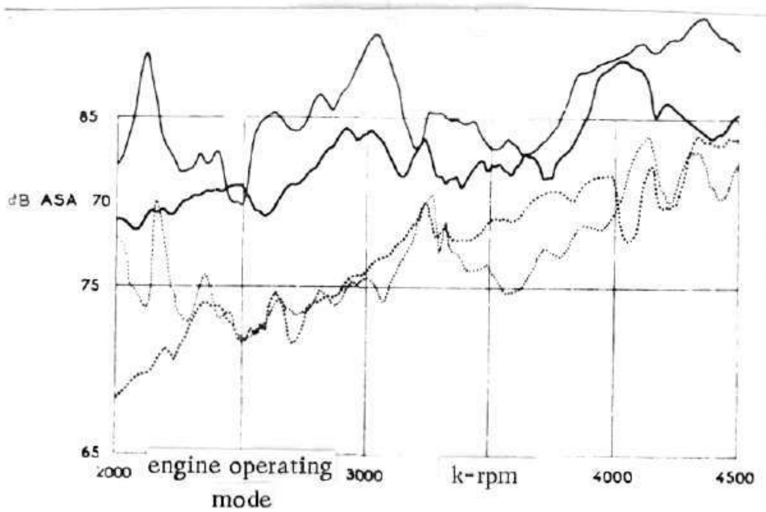


Plate 25. Effect of Load on the Sound Level of a 403 on a Roller Stand. Intake muffler of Figure 16, exhaust noise suppressed: cf. text.

Front seats: ——— full load
...... zero load (neutral position)

Rear seats: ——— full load
...... zero load (neutral position)

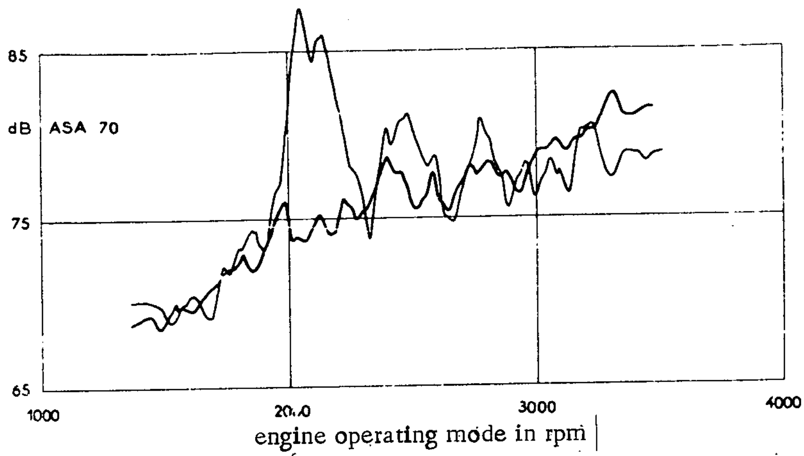


Plate 26. Noise Level Inside the 403 on the Roller Stand. Intake removed to great distance, exhaust noise suppressed, engine driven by the rollers with direct attachment, without fuel and without ignition, slow increase of speed.

——— front seats:
——— rear seats: regardless of valve opening.

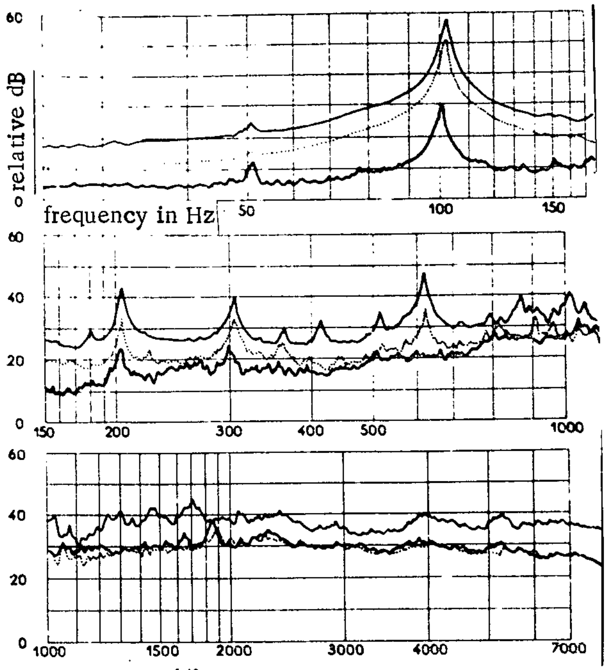


Plate 27. Spectra of Intake Noise Recorded on the Roller Stand on the 403 at N = 3,000 rpm. Motor driven on rollers by direct contact, fuel and ignition cut-off, exhaust noise suppressed, microphone placed 60 cm from the noise source, top down.

________ free intake: full opening
.... empirical muffler: full opening
________ closed valve

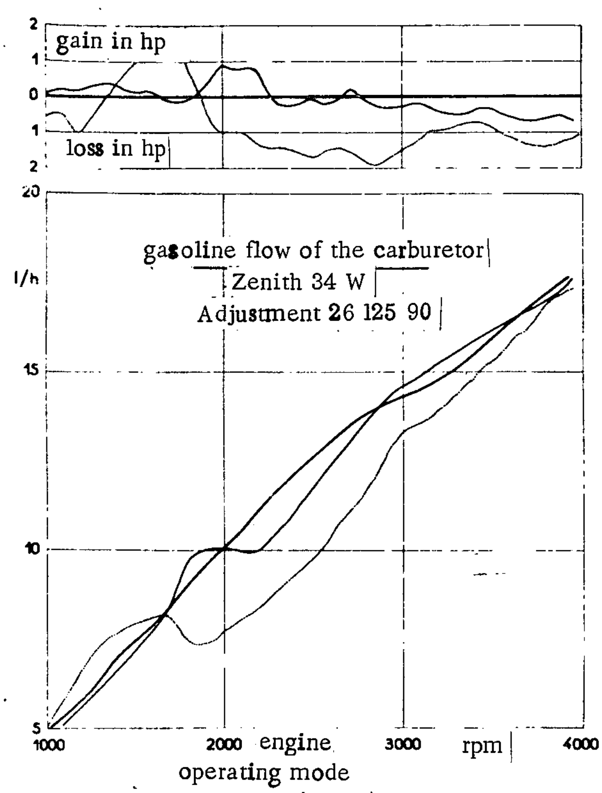


Plate 28. Effect of the Intake Muffler on Carburation, 403 Engine, Full Load. Variation of power with respect to what is measured in free intake (without muffler)

——— free intake;

.... intake muffler of Figure 17;

——— intake muffler of Figure 16.

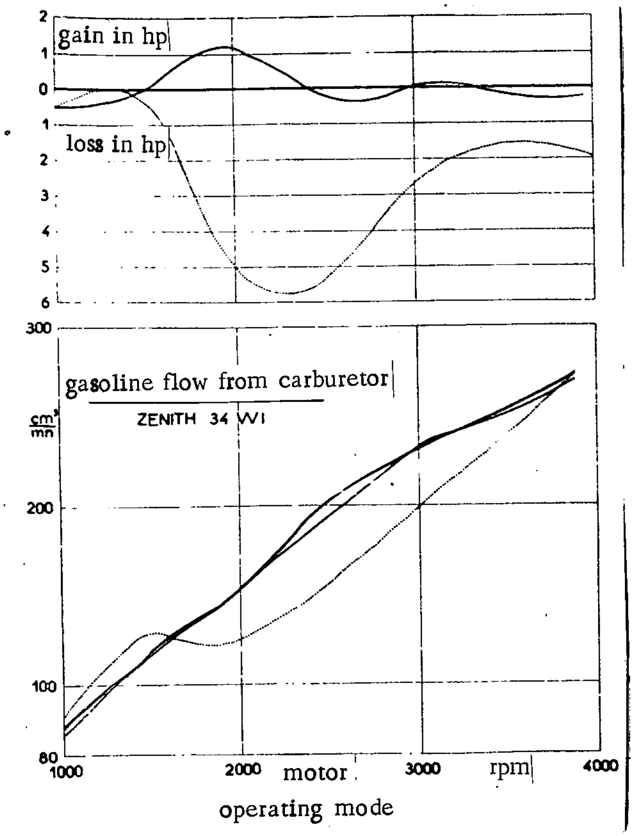


Plate 29. Effect of the Intake Muffler on Carburation (engine 403) Full Load. Variation of power with respect to what is measured in free intake (without muffler).

———— cf. legend for preceding plates;
.... LF, HF, filter with separate high-pass

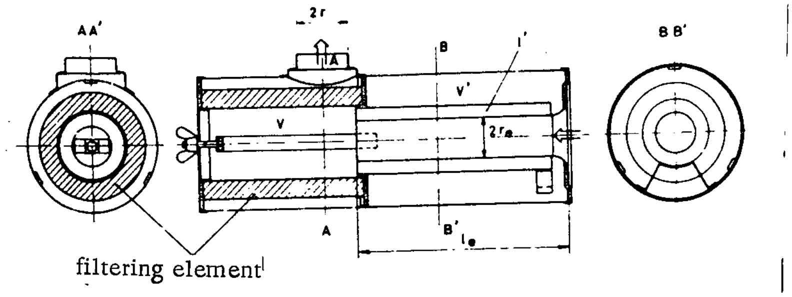


Plate 30. Intake Muffler Consistent with Figure 17.

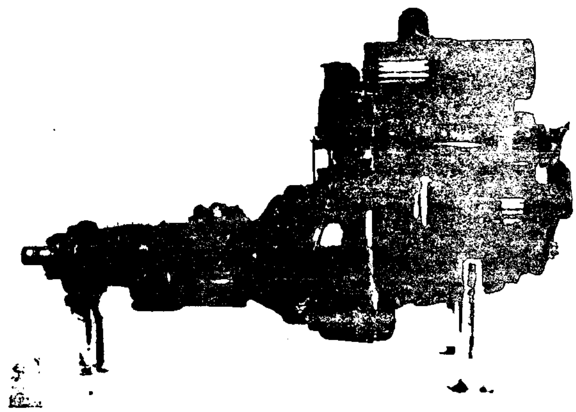


Plate 31. Engine Installation of the Standard Muffler of Plate 30.

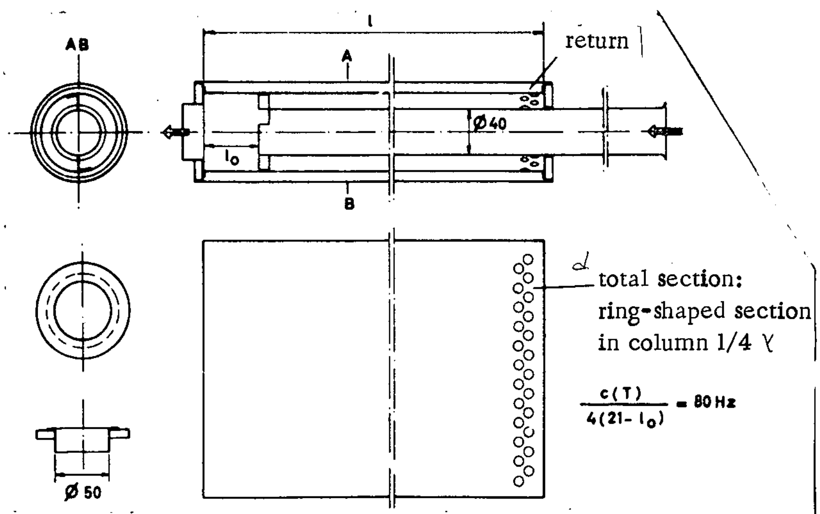


Plate 32. High-Pass of the Special LF, IF, HF Intake Muffler (with Separate High-pass)

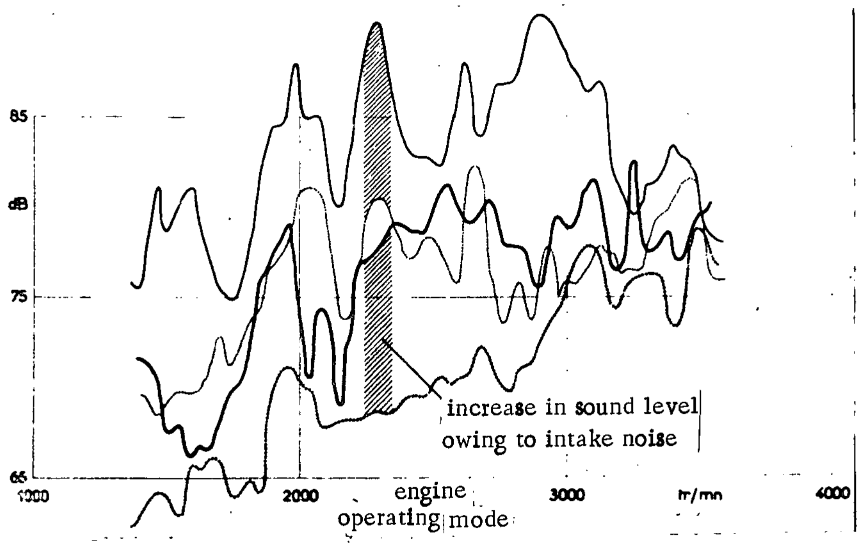


Plate 33. Effect of Intake Noise Inside of the 403 on the Roller Stand. Intake muffler of Figure 16, engine driven by rollers, direct contact.

━━━━━━━━ front seats, full opening
•••••••• front seats, closed valve
———————— back seats, full opening
.... back seats, closed valve

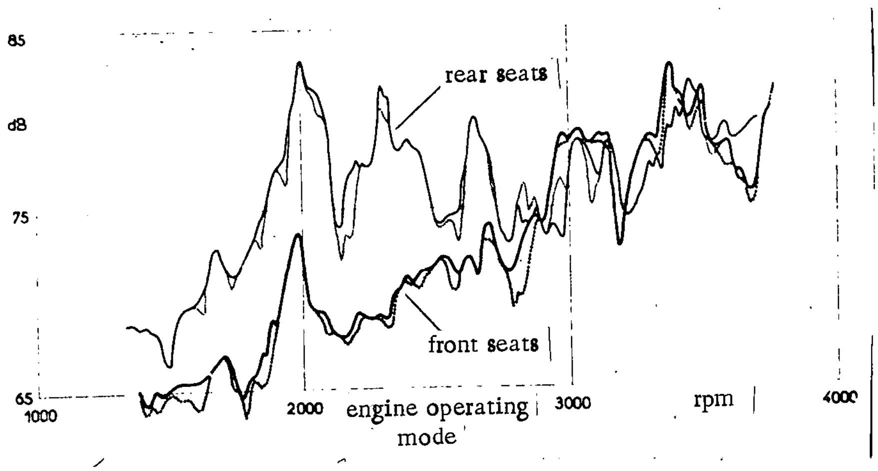


Plate 34. Effect of Intake Noise Inside the 403 on the Roller Stand. Special LF, IF, HF intake muffler, with separate high-pass (prototype). Cf. explanations Plate 33.

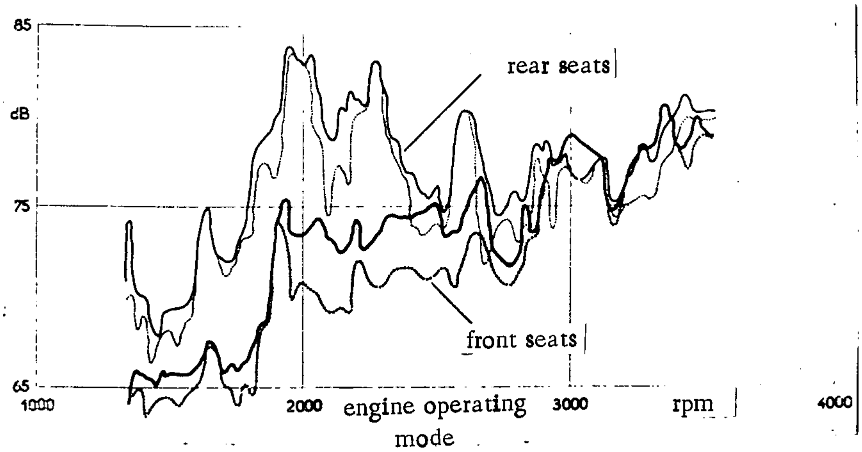


Plate 35. Effect of Intake Noise Inside the 403 on the Roller Stand. Special LF, IF, HF intake muffler, with volume high-pass (Helmholtz resonator), monoblock prototype. Cf. explanations Plate 33.

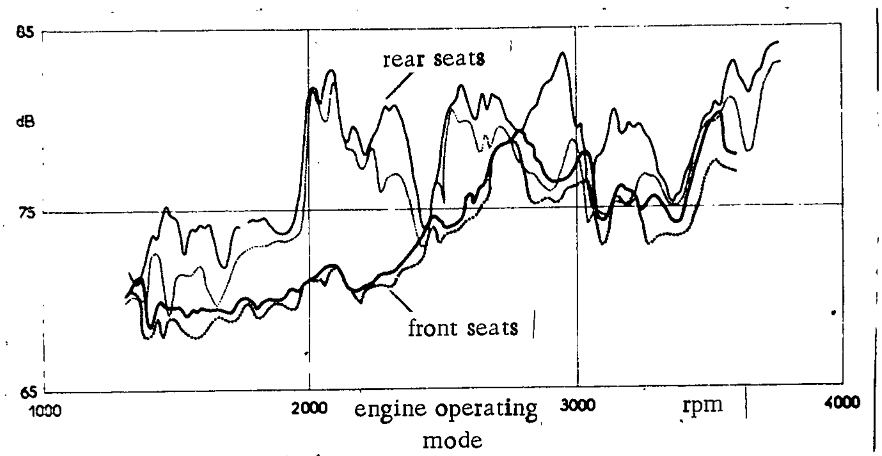


Plate 36. Effect of Intake Noise Inside the 403 on the Roller Stand. Special LF, IF, HF intake muffler, monoblock prototype with rectangular inlet. Cf. explanations Plate 33.

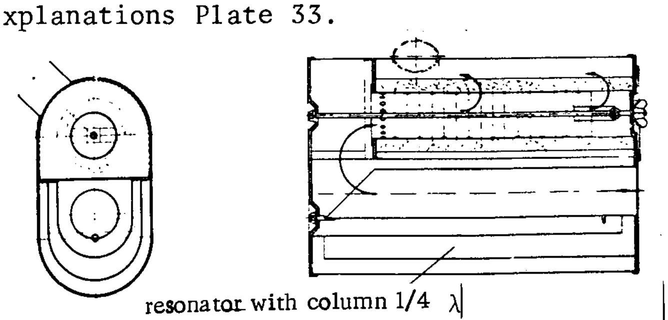


Plate 37. Special LF, IF, HF Intake Muffler, Monoblock with Gas Column.

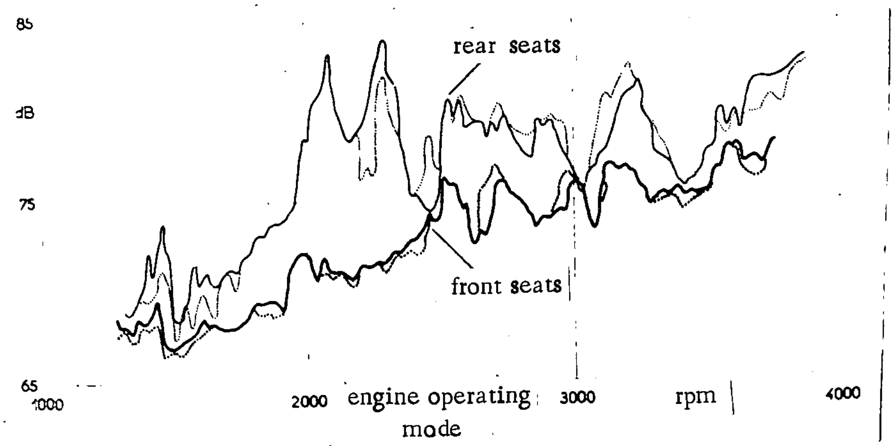


Plate 38. Effect of Intake Noise Inside the 403 on Roller Stand. Special LF, IF, HF, intake muffler, monoblock (prototype with integrated high-pass, with circular inlet). Cf. explanations Plate 33.

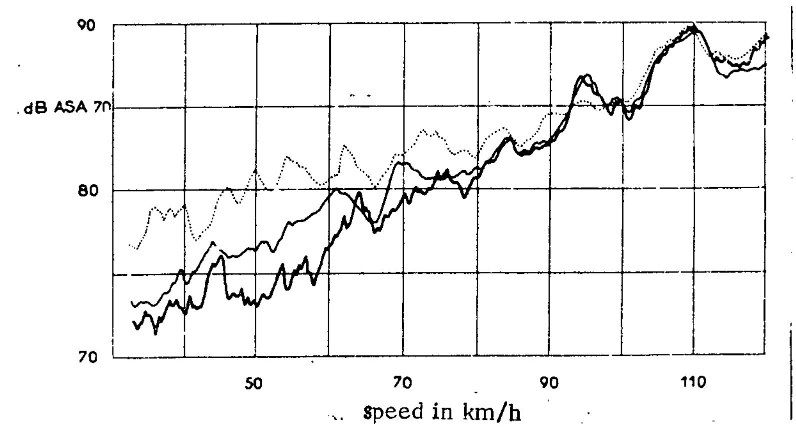


Plate 39. Noise Levels Inside the 403, on the Highway in the Front Seats, in First.

....	intake muffler	of Figure 16: special LF, IF, HF Monoblock	full load
————			
————	low load.		

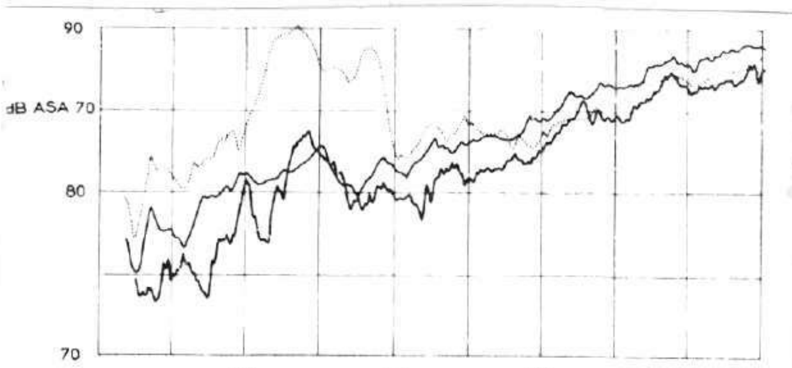


Plate 40. Noise Levels Inside the 403, on the Highway in the Rear Seats, in Fourth.

.... } intake muffler { of Figure 16: special LF, IF, HF Monoblock }, full load.

——— low load.

———

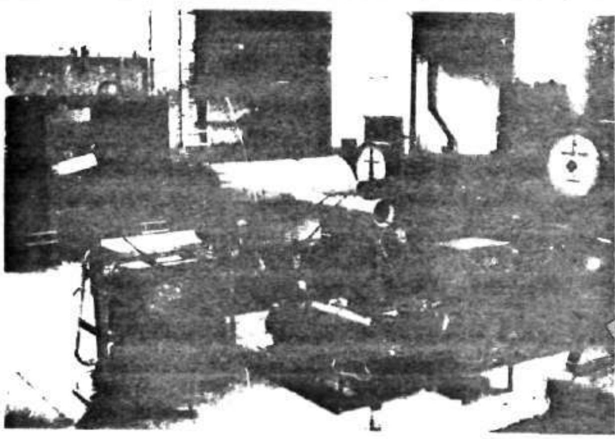


Plate 41. Installation for Measuring Filling in of Piston Engines, with Direct Redoubt.

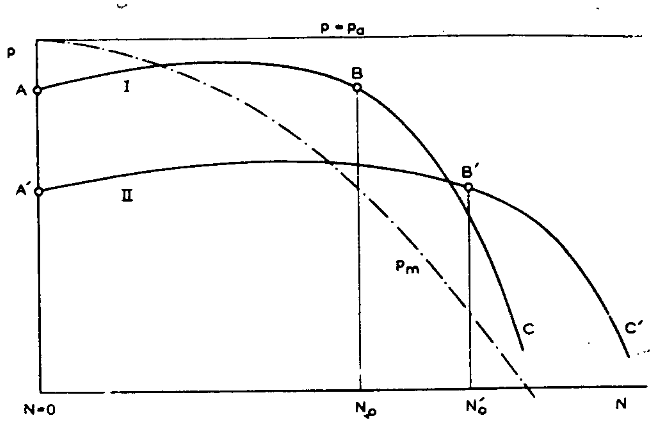


Plate 42. Base Curve of Intake Pressure. α'_{RFA} (curve II) > α_{RFA} (curve I).

AB: A'B': zone of compression. BC: B'C': zone of drop (cf. text).

$N = \frac{N_{SA}}{3}$	h_i	$h_1 = \frac{N_{SA}}{180}$	$h_2 = \frac{N_{SA}}{90}$	$h_3 = \frac{\sim}{60}$	$h_4 = \frac{\sim}{45}$	$h_6 = \frac{\sim}{30}$
	H_i					H_1
$\frac{N_{SA}}{2}$	h_i	$\frac{\sim}{120}$	$\frac{\sim}{60}$	$\frac{\sim}{40}$	$\frac{\sim}{30}$	$\frac{\sim}{20}$
	H_i				H_1	
$\frac{2}{3} N_{SA}$	h_i	$\frac{\sim}{90}$	$\frac{\sim}{45}$	$\frac{\sim}{30}$	$\frac{\sim}{22{,}5}$	$\frac{\sim}{15}$
	H_i			H_1		
N_{SA}	h_i	$\frac{\sim}{60}$	$\frac{\sim}{30}$	$\frac{\sim}{20}$	$\frac{\sim}{15}$	$\frac{\sim}{10}$
	H_i		H_1			H_3
$\frac{3}{2} N_{SA}$	h_i	$\frac{\sim}{40}$	$\frac{\sim}{20}$	$\frac{\sim}{13{,}33}$	$\frac{\sim}{10}$	$\frac{\sim}{6{,}66}$
	H_i				H_3	
$\frac{5}{3} N_{SA}$	h_i	$\frac{\sim}{36}$	$\frac{\sim}{18}$	$\frac{\sim}{12}$	$\frac{\sim}{9}$	$\frac{\sim}{6}$
	H_i					H_5
$2 N_{SA}$	h_i	$\frac{\sim}{30}$	$\frac{\sim}{15}$	$\frac{\sim}{10}$	$\frac{\sim}{7{,}5}$	$\frac{\sim}{5}$
	H_i	H_1		H_3		
$\frac{5}{2} N_{SA}$	h_i	$\frac{\sim}{24}$	$\frac{\sim}{12}$	$\frac{\sim}{8}$	$\frac{\sim}{6}$	$\frac{\sim}{4}$
	H_i				H_5	
$3 N_{SA}$	h_i	$\frac{\sim}{20}$	$\frac{\sim}{10}$	$\frac{\sim}{6{,}66}$	$\frac{\sim}{5}$	$\frac{\sim}{3{,}33}$
	H_i		H_3			
$\frac{10}{3} N_{SA}$	h_i	$\frac{\sim}{18}$	$\frac{\sim}{9}$	$\frac{\sim}{6}$	$\frac{\sim}{4{,}5}$	$\frac{\sim}{3}$
	H_i			H_5		
$5 N_{SA}$	h_i	$\frac{\sim}{12}$	$\frac{\sim}{6}$	$\frac{\sim}{4}$	$\frac{\sim}{3}$	$\frac{\sim}{2}$
	H_i		H_5			
$6 N_{SA}$	h_i	$\frac{\sim}{10}$	$\frac{\sim}{5}$	$\frac{\sim}{3{,}33}$	$\frac{\sim}{2{,}5}$	$\frac{\sim}{1{,}66}$
	H_i	H_3				
$10 N_{SA}$	h_i	$\frac{\sim}{6}$	...	...	...	.
	H_i	H_5				

Plate 43. Acoustic Supercharging. Characteristic operating modes, harmonics of excitation and resonance.

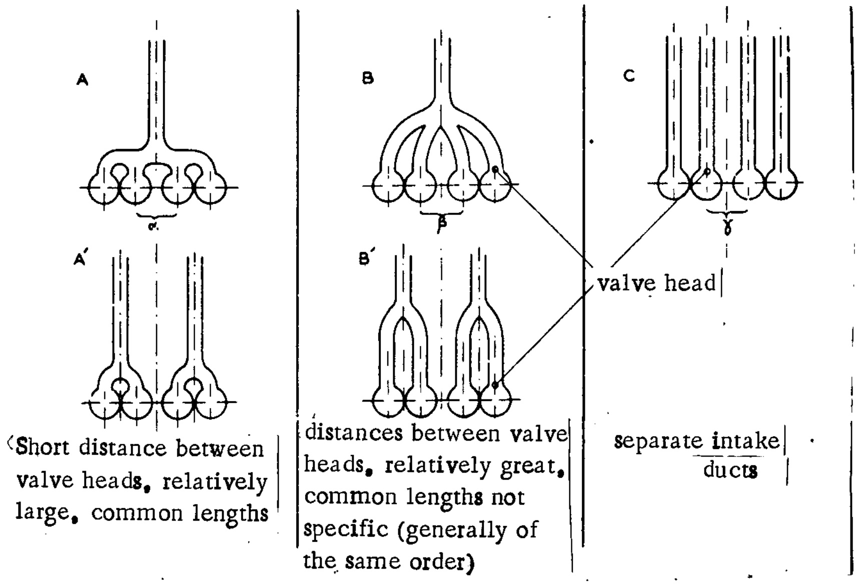


Plate 44. Types of Intake Ducts (4 Cylinders) α, β, γ: Same Types for Two Cylinders,

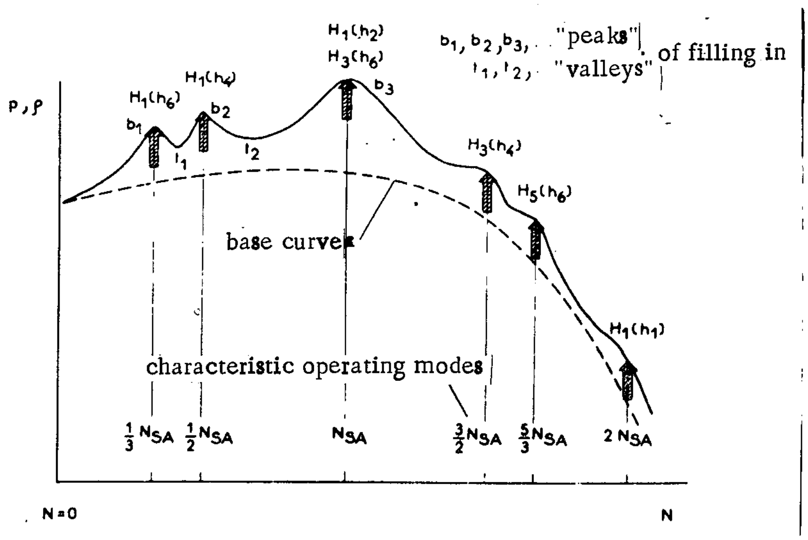


Plate 45. Amplification of the Base Curve by Acoustic Supercharging.

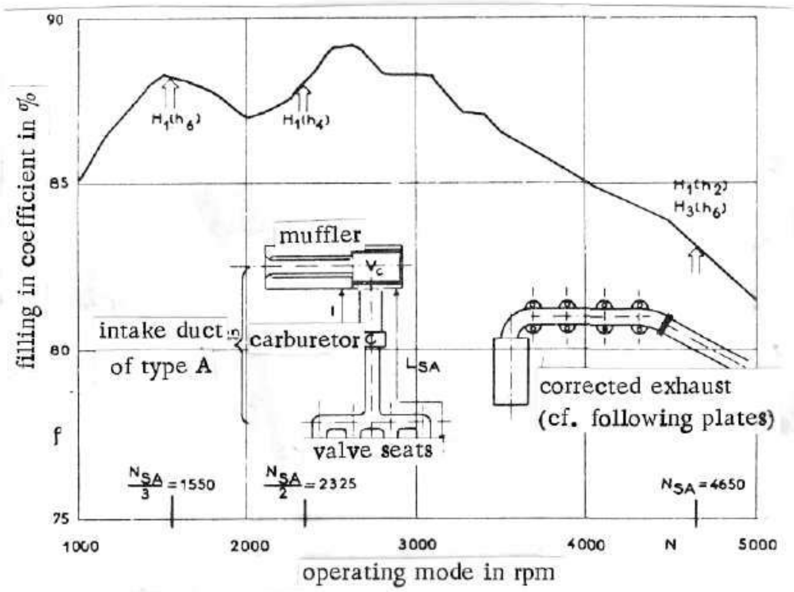


Plate 46. Example of Random Filling in Curve But Devoid of Disturbances Owing to Exhaust.

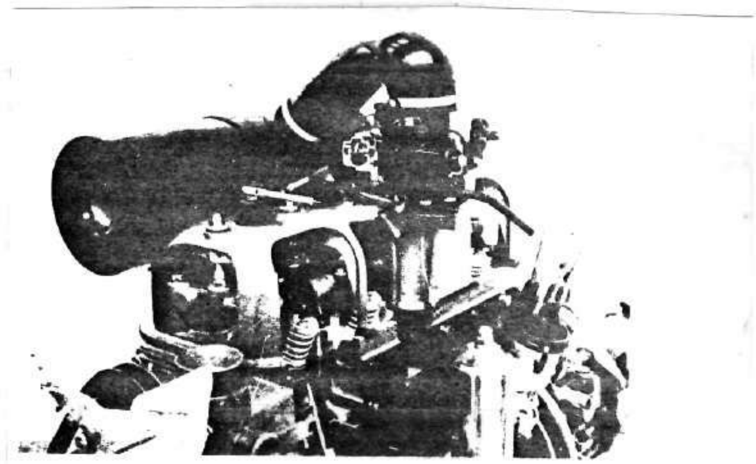


Plate 47. Intake Duct of the Peugeot 403 Engine of Type A.

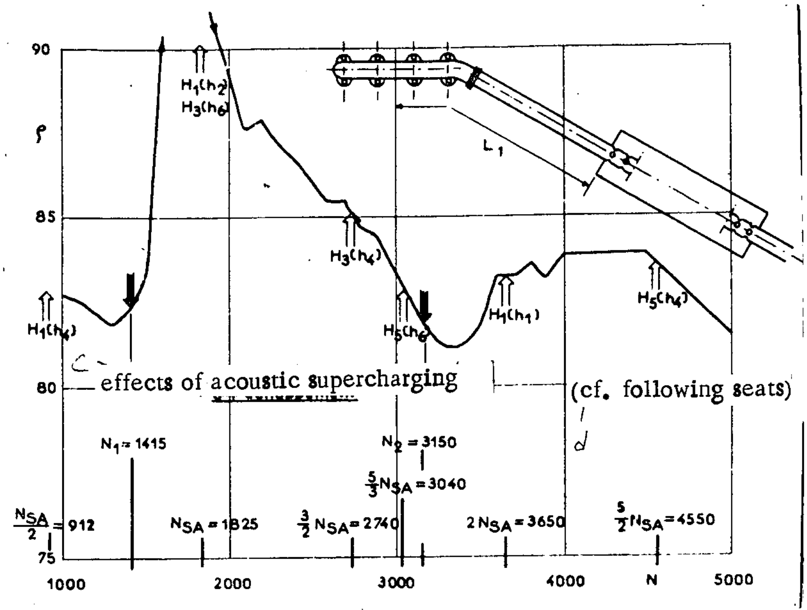


Plate 48. Example of Very Energetic Acoustic Supercharging Possible at Low Operating Modes and Filling in Holes Due to Exhaust.

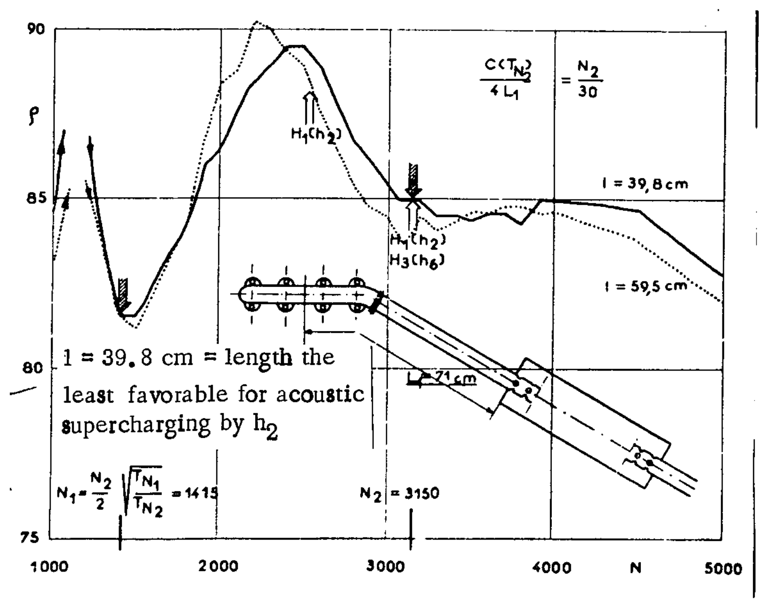


Plate 49. Localization of Disturbances Owing to Exhaust (Effects of Resonance from the First Section of the Pipe: Valves: First Muffler with Simple Volume) on the Stand.

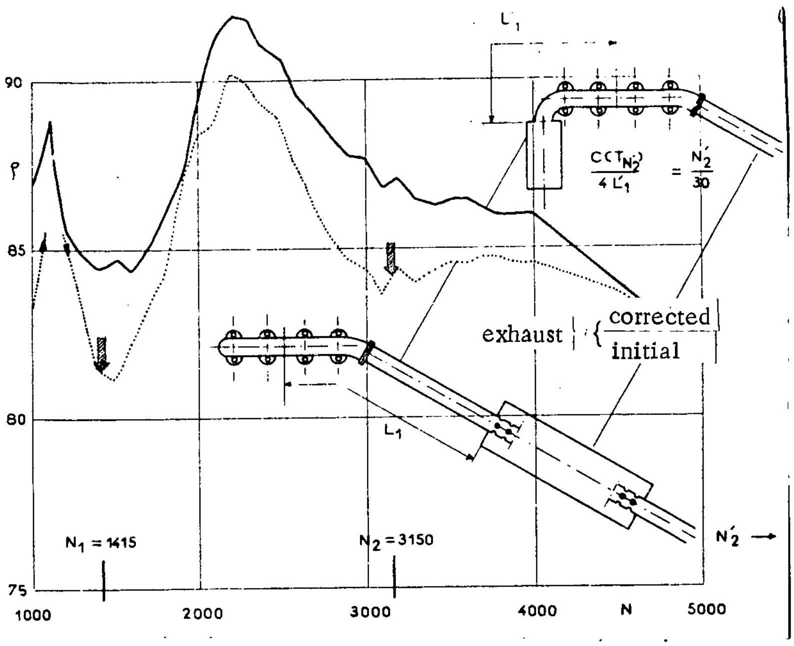


Plate 50. Correction of Exhaust by Reduction of L to the Minimum (L Produced in Series with the Peugeot 404 Engine).

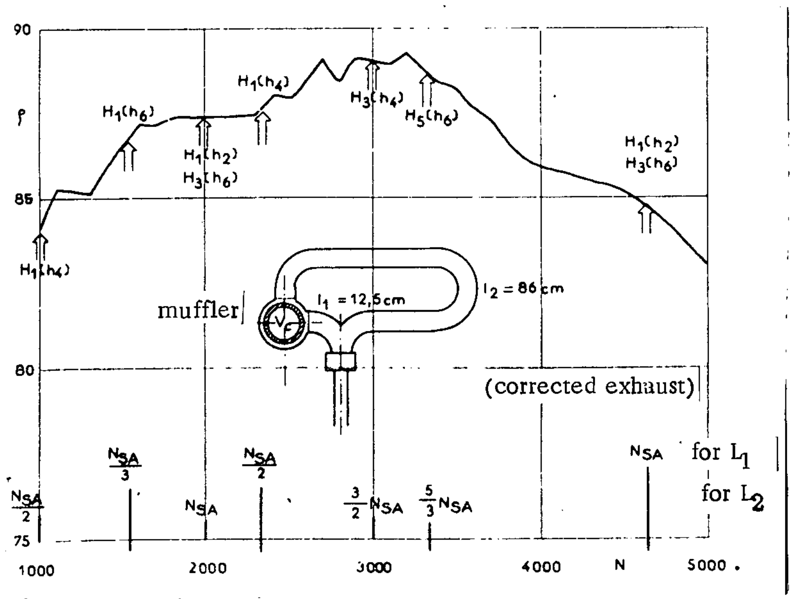


Plate 51. Suppression of Filling in Holes by Careful Superposition of Two Filling in Curves (Special Muffler, with Two Rubber (Durite) Pipes).

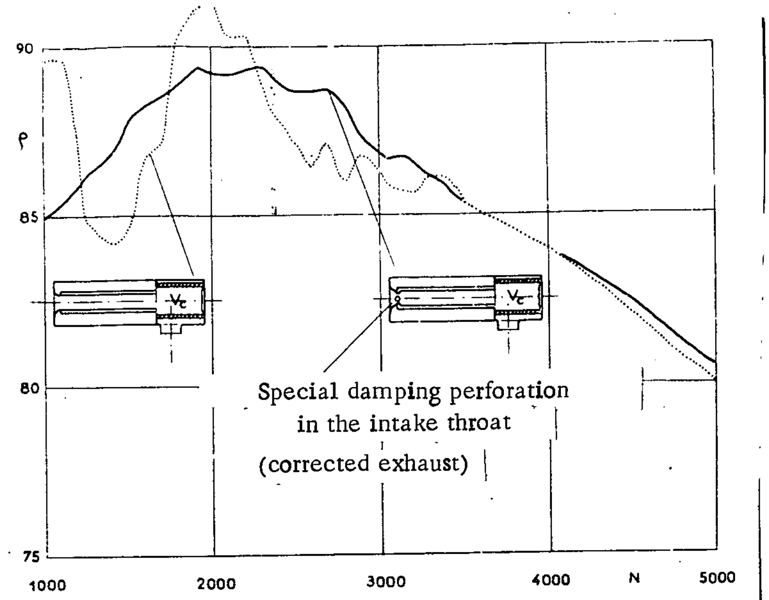


Plate 52. Suppression of Filling in Holes by Optimum Damping of Resonance Phenomenon of L_{SA} (Produced in Series: Peugeot 404).

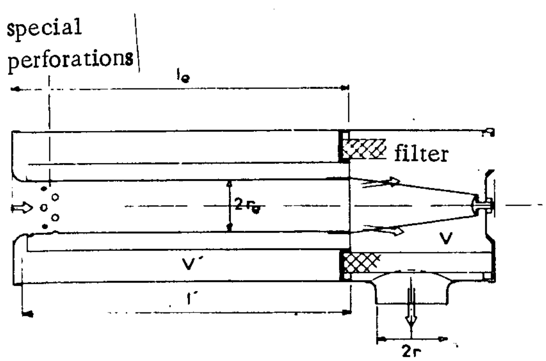


Plate 53. Peugeot 404 Intake Muffler.

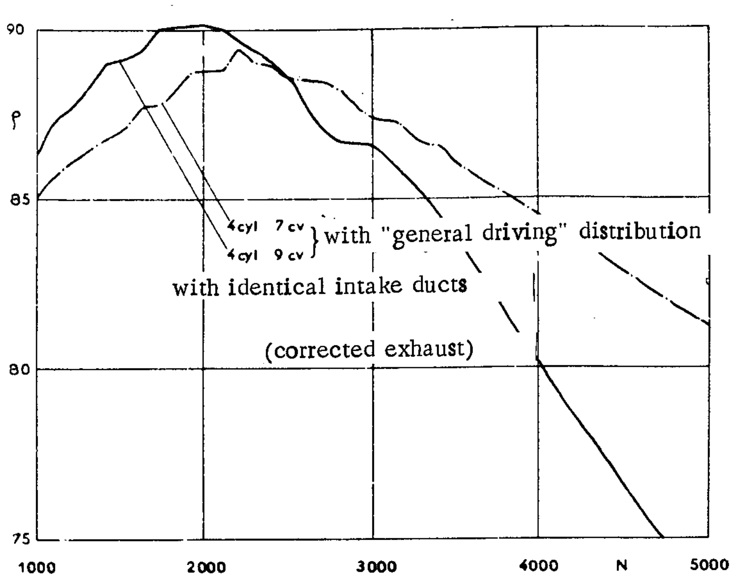


Plate 54. Variation of Filling in as a Function of the Cylinder Capacity Alone (Shape of Optimized "General Driving" Curve).

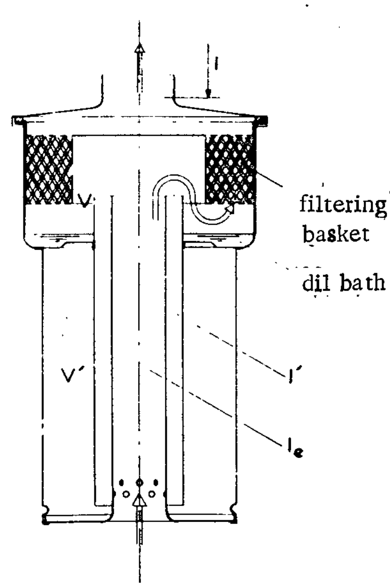


Plate 55. Intake Muffler with Oil Bath for Four Gasoline Cylinders (with Connection to the Carburetor by a Well-Specified Length).

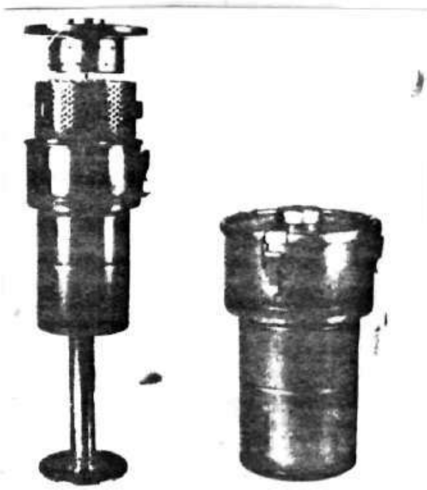


Plate 56. Filter Muffler with Oil Bath for Four Cylinders with Carburetor Branched Off From the Muffler (Plate 53).

Plate 57. Arrangement of the Intake Muffler and Primary Exhaust Muffler. Case of a very short L (cf. text).

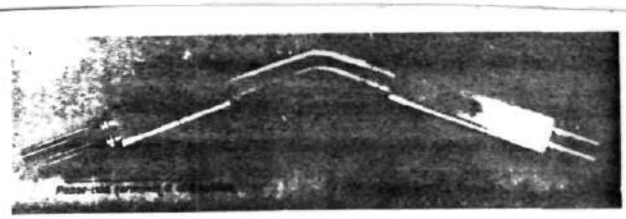


Plate 58. Racing Exhaust for Four Cyclinder with a Length of 1,600 cm^3 (145 hp).

REFERENCES

1. Zeller, W., *Technische Larmabwehr* [Technical Noise Abatement], Krener, Stuttgart, 1954.
2. Waetzmann, E. and F. Noether, *Annalen der Physik*, Vol. 13, 1932.
3. Stewart, G. W. and R. B. Lindsay, *Akustik* [Acoustics], Berlin, 1934.
4. Cremer, L., *Akustik*, Zeitschrift, Vol. 5, 1940.
5. Piening, W., *V.D.I.*, No. 81, 1937.
6. Bentele, *V.D.I.*, Berlin, 1938.
7. Kluse, M., *A.T.Z.*, Vol. 36, 1933.
8. Martin, H., U. Schmidt and W. Willms, *M.T.Z.*, Vol. 2, 1940 and Vol. 3, 1941.
9. Davis, Don D., George M. Stokes, Dewey Moore and George L. Stevens, Report No. 1192, National Advisory Committee for Aeronautics, 1954. Report No. 1192
10. Ney, *Conference CEGOS*, 1956.
11. Billey, L. H., *S.A.E.*, 1950.
12. Muller, L. E., *S.A.E.*, 1959.
13. Serruys, M., *La Machine-Outil Francaise*, No. 136, etc., 1958.
14. Reyl, G., *A.T.A.*, 1954.
15. Wiedeman, A. and M. D. Pugh, *Diesel Power*, 1952.
16. Tutt, F. J. H., *Automobile Engineer*, 1955.
17. Goodger, E. M. and J. Hillsdon, *Automobile Engineer*, 1958.

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